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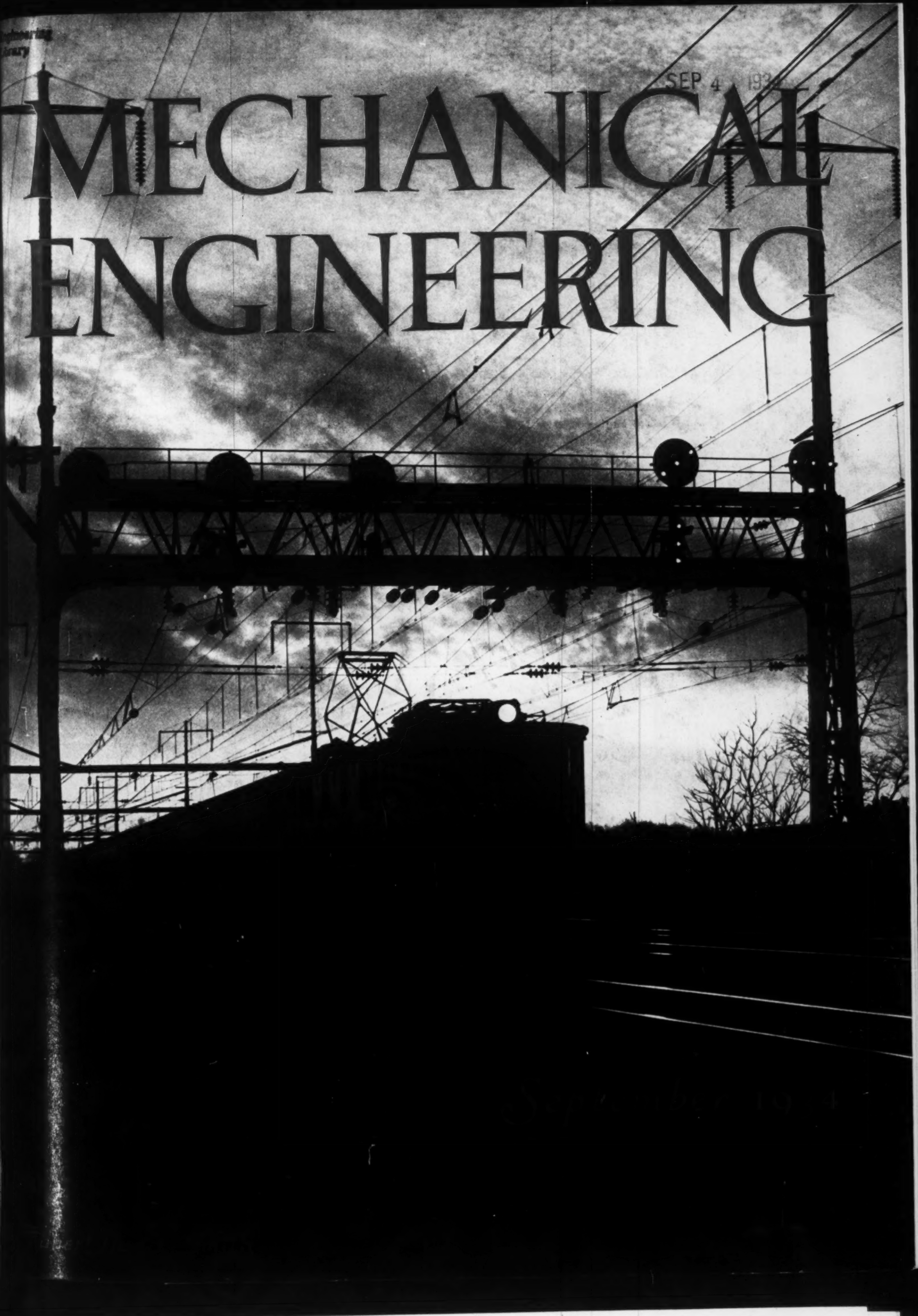

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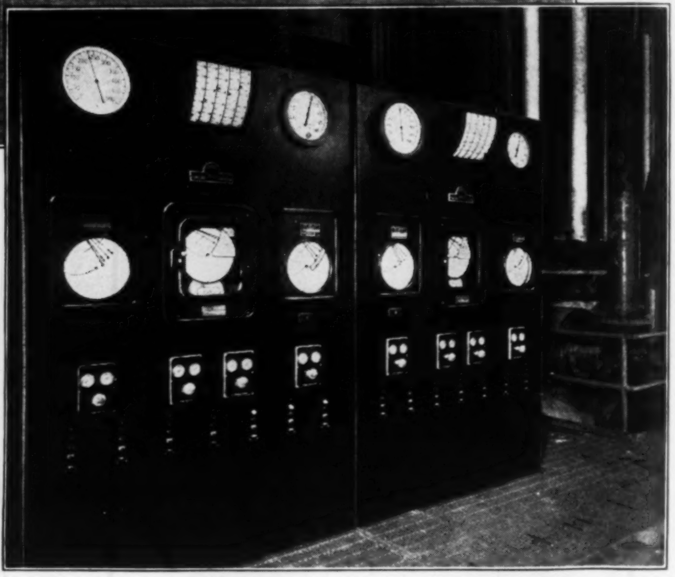

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MECHANICAL ENGINEERING

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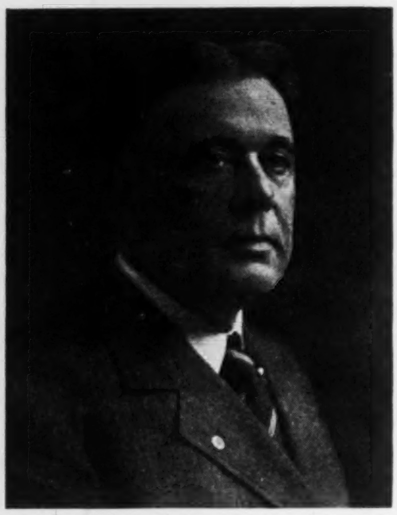
A PROGRAM *and a* REPORT



SEPTEMBER brings plans for the fall and winter period of A.S.M.E. activities, details of which will be announced from time to time. These plans include (1) the extension of the improved type of student-branch organization to the remainder of the United States; (2) the aiding and encouraging of younger members to undertake self-training in technical and cultural subjects and the organizing of local study groups to support this activity; and (3) the building up of present activities looking to the increased prestige of the engineer and greater opportunities for his services.

As a step in giving effectiveness to the tasks that lie ahead, the Council at its recent meeting restated the purposes of the Society. These will be found on page 572 of this issue. For the carrying out of these purposes the Council, during the fall, will present proposals for changes in the Constitution relating to membership grades. The reason for these changes is to increase the standing and prestige of all grades of membership.

The status of those members who have been unable to pay dues has been clarified. The membership list to be issued in December will contain the names of all members in good standing.

But professional activities cannot be undertaken successfully if finances are not sound. I am happy to be able to report that our finances have been put on a sounder basis. The fiscal year, which closes on September 30, promises to show an operating surplus. The debt at the bank has been reduced from $164,000 on October 1, 1933, to about $65,000, partly by disposing of certain of our investments and partly by the sale of certificates of indebtedness to members. Thus, while our income is still greatly reduced, the problem of balancing the budget and carrying on a progressive program of Society activities has become easier.

The Engineering Index has been taken over by a separate corporation and has ceased to become an expense to the Society.

A year of successful accomplishments is bound to result if every member and every committee will embrace the available opportunities to serve the Society and to be served by it.

Paul Doty

An END *to* UNEMPLOYMENT

Unemployment Reserves, Socially Desirable Projects, a Permanent, Efficient Public-Works Administration, and a Nation-Wide System of Employment Offices Constitute Our Lines of Defense

BY RALPH E. FLANDERS[1]

WHY NOT an end to unemployment? There is an astonishing mass of useful work to be done, and there is an abundance of natural resources, organization, and equipment for supporting the workers. Why temporize longer with the problem? Instead of devoting our effort and enthusiasm to subsidiary and unrelated policies, why not squarely face the central enigma, modifying and developing our detailed plans in accord with that face-on view?

At the outset we must recognize the problem of unemployment as being a permanent one. This is true for four reasons:

First, there are some of our citizens who, through faulty inheritance of intellect or character, are unable to render effective service in a society organized for a high standard of living. In a primitive society they would more nearly hold their own; but they must either be carried as a deadweight by any highly organized social group, or must be content to live in the social backwaters, whether of the city or country, until progress is made in solving the special problem which they present.

In the second place, we must recognize that in the adjustments of a growing and improving industrial order there continuously arise conditions under which workers find themselves attached to inefficient businesses or decaying industries. It becomes necessary for them to detach themselves and seek new connections elsewhere. There are thus many unemployed even in good times, since the number of those who are on the move from hopeless jobs to hopeful ones is always large, and the new connection cannot always be made at once. Seasonal unemployment adds to the total.

There is a third type of unemployment, new to us, developed by trades-union policy and social theory through the codes of the NRA. This is the gradual prohibition of gainful occupations for youths. The codes in general set the minimum age at sixteen; but restrictions as to "dangerous" occupations (with the term so liberally construed that few of them are as hazardous as driving a motor car) close many avenues of employment to those under eighteen; and there is no doubt that strong influences are at work looking to the age of twenty-one as the minimum at which idleness ends and self-support begins. The situation is all the more serious from the fact that it has come upon us at a time when appropriations for education have been cut to the quick, and before satisfactory types of education have been developed to fit the needs of these new unemployed.[2]

Finally, we must be resigned to some measurable amount of increase and decrease in business as a whole. The business cycle will continue. The rhythms of industry are as inevitable, and may be as invigorating, as the rise and fall of the ocean tides. These variations need not and must not be so great as to be destructive, as has been the case in the past; but it is foolish to set as our ideal the lifeless level of the Dead Sea. We must therefore plan for repeated periods in which unemployment increases somewhat above the normal resulting from the preceding causes.

The provisions which we set up for relieving unemployment should therefore be continuous, on a considerable scale, and subject to periodical expansion.

THREE TYPES OF DEFENSE AGAINST UNEMPLOYMENT

For these varieties and causes of unemployment we may well consider three types of defense, variously suited to varying conditions. For the moment we will lightly free ourselves of concern as to the cost of financing them.

The first line of defense is the setting up of unemployment reserves, often miscalled "unemployment insurance."

To be morally justifiable and socially safe, unemployment benefits derived from such reserves should be limited in duration, extending for the maximum of eight or ten weeks which would be required to find a new job under ordinary circumstances. The system must be country-wide, in order that socially progressive areas may not be handicapped in competition with backward ones.

In addition, the method of investing these reserves in good times and liquidating them in bad times must be very carefully studied and specified, for on a nation-wide scale the wrong policies would render depressions inevitable and intensify their severity, thus spreading the disease it is intended to cure.

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[1] President, Jones & Lamson Machine Co., Springfield, Vt., nominee for President, 1934–1935, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

From an address delivered at a meeting, New York, N. Y., May 28, 1930, of the Metropolitan Section of The American Society of Mechanical Engineers, and subsequently at a conference of present and past members of the Industrial Advisory Board, Hot Springs, Virginia, June 17, 1934.

[2] The reader will be interested in an article by Dr. M. A. May in the *Yale Review* for March, 1934, entitled "Education for the Unemployed."

For more prolonged unemployment, and for the third class of unemployed youths, a second line of defense is needed. It must be capable of rapid expansion and contraction while maintaining efficiency of operation. It must produce useful results in return for continued subsistence. It must have in it such socially desirable elements as will permit its use for extended periods without weakening the will and ambition of those whom it helps. It must not offer such wages that it will be preferred on this score to private employment. Little more than subsistence for the worker or his dependents should be offered.

More than a generation ago the Harvard philosopher, William James, wrote the most discerning dissertation on the maintenance of peace that has ever been penned. He described in detail the moral elements in warfare—self-sacrifice, discipline, community of purpose, the devotion of individual abilities and possessions to objectives immeasurably higher than those of any individual—and he proclaimed the doctrine that warfare would continue its appeal to the finest and most virile spirits of each new generation unless and until it was replaced by something holding equal values. He demanded, in short, a "moral equivalent of war."

William James proposed as the moral equivalent an organized attack on the obstacles which Nature presents to the progress of man. He saw in his imagination millions of young men, drafted, or enlisting for a service of one or two years, engaged in draining swamps, driving highways through the mountains, assaulting in force the strongholds of Nature wherever she resists the well-being of man.

Is it not remarkable that we have come so nearly to meeting these conditions in the CWA, the CCC, and the FERA? How little more would be required to complete the picture!

There is needed first a careful study and laying out in advance of a multitude of socially desirable projects—such projects as shall slowly transform the physical aspects of our country, from the Atlantic to the Pacific and from Canada to the Gulf, into forms capable of ministering to the highest enjoyment of its citizens. Then some form of permanent organization must be provided. The Engineers' Corps of the army suggests itself. It could readily develop "cadres"—skeleton organizations—capable of indefinite expansion, so as to take care of widely varying numbers of workers. The experience in housing, transportation, and direction of work would be professionally valuable. A modified discipline could be developed—something short of wartime rigidity, but adapted to the building of an esprit de corps, and of such sort as would turn the recruit back into private industry a better man for his own purposes than when he entered the service.

It is time to have done with emergency improvisations in this matter, and to develop from it something not merely useful, but having a content of social regeneration.

But beyond these measures is needed a body of employment, massive and slow moving, perhaps, but more nearly normal in its nature—something from which the transition to private employment will be natural and incidental. Such a resource is to be found in a permanently and efficiently organized Public Works Administration.

So much has been written and so much done with regard to this matter that little needs to be said here. But two observations may properly be made.

To be effective, a Public Works Administration must continue from administration to administration irrespective of political changes. It was perhaps the most serious mistake of the present Administration to throw overboard the body of preparatory work in this field, made ready to its hand. This error destroyed that coordinated action of PWA and NRA on which the best hopes of a managed recovery depended, and left us with little more than normal processes to add to the primary and effective impulse given through the personality of the President and the early effect of the NRA.

The other point to be noted is that this PWA work, while extensively planned ahead, should consist of normal projects of government—local, state, and national—and should be handled through normal construction channels and under normal working conditions. This work should not be the subject of social experimentation. When a worker passes into this employment from private business or vice versa, the change should be unnoticeable. This is necessary if the change is to be made without social disturbance.

A NATIONAL SYSTEM OF EMPLOYMENT OFFICES NEEDED

We have, then, limited monetary relief, unlimited work on socially useful tasks at subsistence wages, and a normally increasing volume of public construction under standard conditions, concentrated in periods of greatest unemployment. These are our three lines of defense. In addition, we need a new instrument of social organization to bring them into correlated effectiveness.

The needed instrument is a national system of employment offices, or a nationally coordinated system of state employment offices. To this institution must go every one who would draw on this unemployment reserve and every one who seeks subsistence employment. It should also serve as the logical (though not exclusive) recruiting station for labor required on great construction works, whether private or public.

Such a system would give us definite information at any given moment as to the amount, location, kind, and duration of unemployment. For no one should be deemed as unemployed who does not register. Registration will be inevitable on the part of the honest and needy worker, for due and respectable relief comes to him through that act. Non-workers who will not apply are pathological or criminal rather than economic problems. They will by this means be readily recognized and should not be numbered among the respectable unemployed. This system will give us, for the first time, a definite knowledge of the size and character of our problem and we can attack its solution by logical processes. Our ignorance hitherto has been fundamental. There are no reliable unemployment statistics in this country. There is no substitute for the practise of gathering them at the point where relief is offered.

Besides serving this overall purpose, the employment-office system will be in position to help the individual man, watching for the shifts in employment which are favorable to his trade or calling, directing him by groups and individually to opportunities for reemployment in private business. If an effective administration can be built up, these offices will serve the employers also. But they must be given no monopoly in general employment. The competition of properly controlled private exchanges must be permitted if the efficiency of public offices is to be maintained.

We have at hand the materials for a real New Era. We have in the President a leader of consummate political skill, devoted to the task of fruitful social reconstruction. We have an industrial organization, for the moment purged of the poison of financial folly, furnished with the tools, the skill, and the experience for building up the material well-being of society, and by no means deaf to the promptings of the social conscience. In the past, this organization, based on the resources of our country, staffed by its capable citizens, energized by science and engineering, has built up general standards of living unknown in any other time or place. This progress need not and must not end.

If politics and business can come to a working agreement for the public good, it will not end.

MUCH OF THE PROGRAM IS UNDER WAY OR BEING FORMED

It is interesting to pause at this point to observe how much of the ideal program outlined is already under way or in process of formation.

The elements of the "moral equivalent of war," as already noted, exist in the CWA, CCC, and FERA. The planning for it is well under way through the studies in progress under the excellent Planning Board of the PWA. The Engineers' Corps of the Army will need little time or special encouragement to set up its operating staff. Finally, so far as one onlooker can judge, there is excellent material for an organizing head in Mr. Harry L. Hopkins.

The least hopeful element is the PWA. In spite of much excellent work done from the administrator down, its undertakings have suffered from an unwise throwing overboard of the preparatory work done in the previous administration, and from an ignoring of the importance of the time factor and the relation of the work to other elements of the Administration's program. There is a constructive or reconstructive job to be done here.

The governmentally supervised system of employment agencies is completely a development of the future, but it is quite definitely in the plans of the Secretary of Labor. This is an undertaking which she is well fitted to organize and administer, and she should be encouraged to do so.

We are well on the way, via these uncoordinated undertakings, to coordinated measures for the relief of unemployment. We are within reach of that goal which our industrial civilization is physically capable of reaching—a job for every man at any time, whether for thousands or for millions—subsistence employment, or better, when private employment fails. Unless our civilization can furnish this employment, it is doomed to a deserved decay. Let us not deceive ourselves in this matter.

BUSINESS IS AN HONORABLE PROFESSION

We now come to the problem which was passed over earlier in this paper—the means of financing so extensive a program. Two elements must be controlled if we are to bring this large-scale and thorough solution within the range of practical policy. We must reduce the volume of unemployment and finance the relief of that which remains. These are tasks of no mean order.

Our first step is to decide whether we shall endeavor to do this by taking better care of our present economic organization, protecting it from disruption, building it up into a state of health and strength, and setting it to work safely at a higher rate than it has ever known before; or, on the other hand, proceed by throttling it, casting it aside, and seeking some other social servant—or Frankenstein—to take its place. Over this question we are vacillating when vacillation is dangerous.

Our first duty is to understand the nature of the profit-supported, credit-money business organization by which we live. Until we have some notion as to its functional activity we can neither decide whether it can be made to suit our purposes, nor can we be sure that our hopeful experiments may not prevent it from serving the ends we seek.

The term "business," which we have been using, is an honorable word. It should be confined in its application to those activities involving land, labor, capital, and management which are essentially wealth-producing, whether in the fields of extraction, manufacture, transportation, distribution, or financing.

There are other activities which are not productive but destructive, and thus do not deserve the honorable name of business. These activities have been due in part to ignorance, in part to wrong standards of social conduct. They are to be found to a certain extent within the structure of business properly so-called in the form of unfair and harmful practises; but their greatest development has been as malignant financial and speculative growths which are for the most part engendered and nourished by individuals who cannot properly be called "business men."

There has been a failure on the part of honest observers of the industrial process to distinguish between true business and these parasitical practises. This failure to distinguish threatens business with kinds and degrees of control which will seriously curtail its ability to produce and distribute wealth, and will thus render hopeless any such social program as we are considering.

The productive capacity of business is such that if it is fostered in its present and future development it can increase the production and distribution of wealth to all classes of society to the disadvantage of none, and at the same time permit the accumulation of ample reserves for any social emergency.

The amount of possible employment is not limited so that it has to be shared. Purchasing power is not a fixed amount so that it has to be haggled over. New employment and new purchasing power can be created by encouraging business enterprise in an expanding production and distribution of goods and services; and this enterprise and this expansion can be so ordered as to avoid the follies of the era of speculative inflation.

It is evident that if business is to perform its normal, productive functions, there must be important changes in attitude and policy on the part of all elements of the social structure.

Labor must be willing to govern its requests by the overall requirements of the industrial process, and not by the immediate self-interest of any restricted group.

Financial interests must refrain from building top-heavy credit and speculative structures, and must be willing to look to true business service for their profit.

Government must distinguish between business in its proper sense and destructive practises which are unworthy of the name, giving the former adequate freedom and active encouragement, while it restricts the latter. The methods of restriction must, therefore, be considered with great care. Above all, neither legislation nor administration policy should include elements which are unnecessary to the stated purpose, and which indicate ulterior and unstated purposes hostile to the profit system; and care must be taken to avoid the appointment of officials having discretionary power who are hostile to the profit system, and therefore incapable of assisting it to attain its full social usefulness.

Government must provide a sound monetary and credit system of reasonably steady purchasing power—these two characteristics not necessarily being contradictory; thus will business and agricultural operations[3] be steadied, and financing for improvement and expansion be assured. It must encourage the free flow of savings into productive investment, instead of putting the whole process under suspicion.

The nature of our credit-money system is such that the normal means of increasing purchasing power is by an increase in bank borrowing. Available credit has been multiplied by governmental policy. But there must also be a willingness to borrow on the part of productive borrowers. Here the government's policies are seriously in question, in that they have generated a wide-spread distrust as to the permanence of our system—a distrust inevitably resulting from the legislative and appointive errors already mentioned.

Safety, as well as volume, of credit-money is essentially important. Safety—with no danger of subsequent deflation—would seem to be assured if we confine credit expansion to the financing of current operations of production and distribution, leaving capital expansion dependent on savings alone.

If business itself is to serve the social purposes I have just outlined, and serve its own interests in so doing, it must discipline itself, or submit to being disciplined, in several particulars, among which are the following:

Business must be willing to outlaw those forms of competition which are in the long run socially destructive, even though immediately profitable.

Business must not be shortsighted in its distribution of the returns from an expanding industry, but must give due shares to labor, to the consumer, to management, and to capital.

Business management must be content with business profits, and not make its principal interest the search for illusory and unsocial speculative gains.

Business must not seek to protect itself unduly, but must be willing to undergo the dangers of such a degree of competitive freedom as will insure the orderly and continuous application of improved processes to lowering prices for the general good. Neither private profit nor general well-being will be served in the long run by attempting unduly to safeguard business operation and profit, even for the sake of stabilizing employment. Some measure of uncertainty, some percentage of mortality, is the price of social advance. But, while business retains its dangers, the worker must be safeguarded.

Business must cheerfully accept proper taxation for socially desirable purposes, and be willing in other ways to submit to that minimum of government control which the general interest requires.

Finally, business must develop a realistic social philosophy. Its purposes must be crystal clear, its policies humanly useful.

COOPERATION BETWEEN BUSINESS AND GOVERNMENT

Let us further consider this lack of cooperation between business and government—this misunderstanding of our profit-supported, credit-money economy. It cannot be denied that profit itself is under suspicion.

It is profit from legitimate business as already defined, that is socially desirable. The immense profits of the 1928–1929 period were not earned from business but were derived from speculative inflation. They were, indeed, not profits at all, being mere borrowings on the future, based on bank indebtedness which had to be made good later not always by the original borrower, and in the payment of that indebtedness purchasing power was disastrously extinguished.[4] "Profit," like "business," should be so defined that it becomes an honorable term, and a rational one as well.

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[3] For the relations of agriculture to the industrial program see the author's paper "Raising the Farm Standard of Living," before the American Association for the Advancement of Science, Dec. 30, 1933, and published in MECHANICAL ENGINEERING for March, 1934, pp. 135–138.

Such profit must be actively encouraged—not merely tolerated. The revival of the sick industries, the reemployment of the unemployed, depends on the reappearance of generous profits. Only so can those unfortunate workers who are dependent on the capital-goods industries, on luxury and durable-goods manufacture, and on the service occupations hope to regain a foothold in our industrial society.

Shallow thinking has led to the doctrine that profit dries up activity; and this erroneous doctrine has hampered recovery. Profits, even when put in the bank, do not disappear from purchasing power. They instantly revitalize the circulating stream.

There are but three ways in which profits can disappear from circulation: by hoarding in the form of currency; by expatriation through unrepayable foreign loans or otherwise; or by extinguishment in the banking system when bank loans are being paid off faster than new ones are being made. We have been through all of these disasters in the past four years. So far as concerns the mechanics of the situation, these conditions *were* the disaster itself. The danger of renewing the two first is remote. The third becomes inevitable only after a period of inflation, and that we must beware of entering upon, for therein lay the causes of the miseries from which we are recovering.

No! Profit is safe, and profit is necessary. We are so organized that, if we guard the three channels by which it may escape, it cannot fail to flow through the whole industrial structure and support in growing comfort the multitudes dependent on it; and in so doing it will inevitably tend to a broader distribution that is on a higher level—a distribution of plenty, not of need; and it will reach every useful member of our industrial society. We must turn our hand to this constructive process, instead of concentrating governmental support on labor policies, many of which we may view with sympathy, but yet with the realization that they have no power in themselves to add a yard of cloth or a single automobile tire to the consumption standard of the American worker.

Finally, profit is necessary for the social program. If encouraged, it can grow beyond any previous experience, so far as its total is concerned, even though the spectacular accumulations from non-business operations are prevented. These more widely distributed returns can easily support the taxation needed for such a radical development as we have been discussing, and for others left undiscussed, such as old-age pensions. If the hope of profit is destroyed, or if real profits are not allowed to exceed our best previous performance, such programs are nothing but useless waste paper and empty breath. For they have no other source of support in the heavens above, the earth beneath, or the waters that are under the earth.[5]

BUSINESS NEEDS POLITICS—POLITICS NEEDS BUSINESS

For the sake of the common man, business and politics must come into new relations with each other. This is the conclusion of our study of the unemployment problem.

Business needs politics. For, in the end, politics rules. The controlling forces are not industrial, or financial, but social; and with these social forces it is the task of the politician to deal. As to the strength of these forces, or the skill with which they may be manipulated, there should be no remaining doubt in the mind of one who has observed the history of the past two years in this country, or of the last fifteen years in Europe. Politics rules —but its rule may as easily spell disaster as progress.

Politics and politicians need the help of constructive and productive business. For government is a clumsy and hopeless instrument for the production and distribution of material goods and service. It is business that provides goods and distributes them through employment. Without the profitable operation of business there is not the faint shadow of a chance that the populations on whose support the politician depends will have such satisfaction in life as to make them content with the leaders whom they have chosen. Without that satisfaction there is for the politician no alternative save desperate leadership in the descending spiral of social destruction—an act not of politics but of lowest demagoguery. "Politics," like "business" and "profit," is an honorable term, and has its standards of conduct.

This essential alliance must be an open one. Its ends, while personally profitable, must be of such evident social worth that business can bring its personal and financial support to a candidate, to a policy, or to a party, without concealment and without shame.

We have at hand the materials for a real New Era. We have in the President a leader of consummate political skill, devoted to the task of fruitful social reconstruction. We have an industrial organization, for the moment purged of the poison of financial folly, furnished with the tools, the skill, and the experience for building up the material well-being of society, and by no means deaf to the promptings of the social conscience. In the past this organization, based on the resources of our country, staffed by its capable citizens, energized by science and engineering, has built up general standards of living unknown in any other time or place. This progress need not and must not end.

If politics and business can come to a working agreement for the public good, it will not end.

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[4] No incident of the depression was more significant than the total disappearance from purchasing power of nearly $1,000,000,000 of soldiers' bonus in two or three weeks in the spring of 1931. It all went to the paying of bank indebtedness. The lesson to be drawn is twofold. It reveals the insufficiency of the "purchasing power theory," on which much of the recovery program is based. And it points to the danger of applying bank credit to capital expansion, or to long-term instalment buying. Such debts must be settled under deflationary conditions, and purchasing power is extinguished in the process, as would not be the case if these activities were financed from savings.

[5] The complementary effect, which I appear not to have emphasized sufficiently, is that the preceding policies will greatly reduce unemployment. This decrease, plus the increase in taxable profits, brings the program within the range of practicality.

The First Year of INDUSTRIAL MANAGEMENT *Under the* NRA

By J. O. McKINSEY[1]

To PRESENT an appraisal of such a program as the NRA, it is necessary to have some standards of measurement just as an engineer who is to appraise equipment must have some standard by which to evaluate it. In the present appraisal of the effect of the NRA on industry in this country, the following standards have been used:

(1) To what extent has the NRA accomplished the primary purpose for which it was established?

(2) To what extent has it produced beneficial results other than those contemplated directly by the NIRA?

(3) To what extent has it produced harmful effects in achieving its primary objectives?

FACTORS THAT HAVE AFFECTED THE NRA PROGRAM

Before attempting to present an appraisal in terms of these standards, it is well that we consider some of the factors which have affected the operations of the NRA.

The major factors which have affected the NRA program have been the following:

(1) The program was initiated during a period of serious depression and was intended primarily as a means of meeting an emergency rather than as a means of formulating a long-time program. The law was passed in haste. It is indefinite and consists largely of general grants of authority to the President. Those to whom the President delegated the responsibility for administering the law had to establish regulations for its administration quickly and while operating under great pressure. They did not have time to study the problems with which they had to deal to any great degree and were not able to conduct the research necessary to formulate sound decisions on these problems.

(2) Neither the Government nor industry had personnel trained in the formulation and operation of such a program. It was necessary for the Government to recruit a large staff quickly, and it had to choose from those who were willing to serve. As a consequence, some of those selected were not well qualified for the performance of the type of responsibilities required in the operation of this law. The men placed in executive positions came from various walks of life and had widely different experiences. Consequently, they did not have a common point of view with reference to the program, and there was not sufficient time to develop coordinated thinking on the part of these various employees. This led to a lack of uniformity with reference to both policies and practises. Since a large number of the employees chosen were not well adapted for this type of work, the turnover in the office of the administrator was large, and this resulted in a further lack of continuity of policy.

(3) Those responsible for the administration of the act were immediately subjected to very intensive pressure from the following groups:

(*a*) Those interested primarily in the political consequences of the program
(*b*) Those interested in economic and social reform, or, at least, change
(*c*) Those interested in increasing the influence and power of labor
(*d*) Those interested in promoting the welfare of the capitalistic system.

Since the Administration did not have any well-defined policies, it was especially susceptible to the influences of these various groups, and, consciously or unconsciously, those who were responsible for the administration of the NRA program were influenced by these groups in varying degrees from time to time. It is impossible to realize the extent and effectiveness of these influences unless one has spent a certain period of time in the atmosphere of the national capital and realizes the insidious way in which the psychology of the mob affects the thinking of the individual. One cannot come in daily contact with those who are enthusiastically supporting a certain point of view without being affected to some degree by their method of thinking. As a consequence, some of those who were influential in the councils of the NRA came to believe that the "R" stood for "Reform" rather than "Recovery."

(4) All of the parties at interest, including the politicians, the labor leaders, and the industrial leaders, were unanimous in declaring that action should be taken quickly. It is probable that this is the only question with reference to which all agreed. As a consequence, an attempt was made to formulate many codes quickly. This led to the adoption of many ill-advised and inconsistent provisions in the early codes.

(5) The NRA in its eagerness to secure results quickly assumed extensive authority and in the exercise of this authority performed three major types of functions, namely, legislative, judicial, and executive. Moreover, the administrator personally performed all these functions and delegated little authority to his assistants. He sought within a six months' period to accomplish by fiat what industry in this country had not been able to accomplish as a result of study and experience within three centuries. As explained later, in my opinion the functions of the administrator should be administrative only.

THE EXTENT TO WHICH THE NRA HAS ACHIEVED ITS OBJECTIVES

With this background, we may safely proceed to consider the extent to which the NRA has been able to achieve its major objectives. It is somewhat difficult to determine accurately

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[1] President, J. O. McKinsey & Co., Chicago, Ill.

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the primary objectives of the program since the law itself is indefinite, and those who have been interested in its operation have varying opinions as to what it was intended to achieve. If we examine the statements of the President with reference to the law and consider the discussion of the proposed law by the congressional leaders at the time of its adoption, we are led to the conclusion that the primary purpose of the law was to aid in stopping deflation and in promoting industrial recovery. It is also quite evident that its sponsors thought that this could be accomplished primarily by (1) increasing employment and wages, and (2) eliminating unfair trade practises which led to price cutting.

The major objective of the sponsors was to achieve the increase of employment and wages. The elimination of unfair competition was a secondary consideration with most of them.

In the achievement of these objectives, the NRA in the early stages was influenced by certain policies which, in my opinion, are fallacious. These are as follows:

(1) It assumed that purchasing power as a whole could be increased by raising wages in advance of increased production. Economists differ with reference to the effect of increasing wages on purchasing power, but the danger which can result from carrying this assumption to its logical extreme is apparent. It is not necessary to understand even the elements of economics to realize that an economic paradise cannot be brought about by increasing *money* wages.

(2) It was assumed that as payrolls increased all industries would be benefited simultaneously. A very little knowledge of human nature teaches us that an increase in the purchasing power of wage earners results first in an increased demand for a certain type of consumer goods. As this purchasing power continues to increase, this demand spreads to other types of consumer goods and ultimately it will lead to a demand for machinery, equipment, and other capital goods; but the process under conditions which existed in 1933 is a slow one, and, in the meantime, the effect of increased wages to the extent that this brings about increased purchasing power affects different industries in varying degrees. As a consequence, the application of the method adopted by the NRA led to many maladjustments among industries.

(3) It was assumed that labor costs represented approximately the same proportion of the total cost of production and distribution in different lines of industry; at least the actions of the administrator indicated the acceptance of this assumption. We know, of course, that this is not true, since labor may represent only 10 per cent of the cost of some products and as high as 90 per cent of the cost of other products. As a consequence, an increase of wages on a percentage basis had widely varying effects on different types of industry.

Because of the apparent acceptance of these assumptions, the NRA program led to many maladjustments. These maladjustments were further aggravated by the fact that some industries were brought under codes more quickly than others, and the provisions in the different codes varied to a considerable degree from industry to industry.

Despite all these factors, the NRA did achieve the primary objectives of the act to some degree. It is self-evident that the adoption of the codes resulted in some increase in employment and some increase in wages. The adoption of these codes did not bring about as extensive reemployment as the Administration had expected and prophesied. It is impossible to secure accurate figures as to the number of the unemployed at the time of the establishment of the NRA. It seems fair to assume, however, that this number was not less than 12,000,000. It is also difficult to determine the number reemployed, but on the basis of the available data, it seems fair to assume that between 3,000,000 and 4,000,000 have been reemployed. On the basis of these data, it seems that since the passage of the NIRA, 25 to $33^1/_3$ per cent of the unemployed have secured employment. It is not possible, of course, to determine to what extent this reemployment has resulted from the operation of the codes and to what extent from improvement in business conditions due to other causes.

It is also apparent to one who is familiar with the operations of a number of industries during the past year that the adoption of the codes has resulted in stabilizing prices to a considerable degree and to increasing prices in many cases, because of the elimination of bad competitive practises or the establishment of uniform prices through the open-price list, or other means. It is again impossible to determine the extent to which prices have been increased through the operation of the codes. This task is made even more difficult because of the changes in the monetary system and the possible effect of these upon prices.

It seems reasonable to say, however, that the NRA has achieved in some measure its primary objectives, namely, the increase of employment and the increase of prices. Although it has not accomplished as much as its sponsors anticipated, it must be realized that it took longer to procure the adoption of codes than was originally contemplated, and that many industries have been operating under the codes only a few months. It is not yet possible to determine the ultimate effect of the codes upon wages and prices. It is also not possible as yet to determine the extent to which the increase in wages and prices will affect industrial recovery.

COMMENDABLE COLLATERAL RESULTS OF NRA

We may now consider briefly the extent to which the NRA has produced effects other than those originally contemplated. We have learned by experience that laws which are hastily conceived and aggressively administered may often produce effects which are quite different from those contemplated by their sponsors. This has been quite true in the case of the NIRA. It is impossible at this time to determine all of the various effects of this act and the ultimate result of these. It is sufficient for present purposes to mention the following:

(1) The adoption and the enforcement of the codes have brought the leaders of each industry together, and this has resulted in these leaders thinking about the major problems of the industry more intensely and more intelligently. Many of these leaders have been stimulated by these contacts to think of many problems of which they were either not aware or did not believe sufficiently important to merit serious consideration. These leaders have also learned much that will be helpful to them in the discussion of their common problems with competitors. It seems incontrovertible that the efficiency of these executives has been increased by these contacts.

(2) Industrial leaders have been made aware of the importance of considering the needs and desires of labor and of recognizing the significance of the public interest in private enterprise. We are not here attempting to determine the justice of the demands of labor or the extent to which Government should exercise control of industrial activities. We are only emphasizing that, regardless of the NRA, it is inevitable that these two parties will take an increasing interest in the management of industry in the future and that industrial leaders will be more effective if they recognize this interest and deal with it intelligently. It is inevitable that in the development of civilization, regardless of the form of government, the millions of workers will secure an increasing influence over the control of the conditions which affect their happiness and prosperity.

If industrial leaders do not recognize this fact, they will encourage the acquirement of greater rather than lesser control on the part of labor. It also seems apparent that in the development of large-scale industry, it is inevitable that the public will exercise an increasing influence in protecting the rights to which it believes it is entitled. The wise industrial leader will recognize this fact so that he may deal with this problem more effectively. In so far as the operation of the NRA has taught industrial leaders to recognize these conditions, it has been helpful in the development of our economic organization.

(3) It has promoted social welfare in some important ways. For example, it has resulted in the elimination of child labor, the decrease of hours of labor for women, and the improvement of working conditions in many cases. None can deny that in so far as the NIRA has aided in the achievement of these ends, it has been beneficial to industry in the long run, as well as beneficial in promoting social welfare.

HARMFUL EFFECTS OF THE NRA

Now we come to a consideration of the extent to which the NRA has produced harmful effects in achieving its primary objectives. Time will not permit a discussion of these in detail. Therefore, only a few of major significance will be mentioned.

(1) The enforcement of Section 7(*A*) of the NIRA has promoted conflicts between labor and capital. Union labor has seen in this section of the law an opportunity for it to strengthen its position quickly. It has not been slow to avail itself of this opportunity. Labor leaders in seeking to enforce what they conceive to be labor's rights under this act have made use of the methods employed by labor since time immemorial, namely, the demand for collective bargaining through union leaders and the threat of strike. As a consequence, there have been numerous strikes and threats of many others. The Government has exercised its influence extensively to prevent strikes in the major industries. The extent to which it will be successful in preventing these in the future, we cannot now foresee. In any case, a considerable amount of ill feeling between labor and capital has been engendered as a result of the NIRA, and the end of the present situation is not now in sight.

(2) There have been many maladjustments between different businesses and between different industries because of the inequalities resulting from the attempt to enforce quickly codes which have been prepared hurriedly. Some members of an industry have been adversely affected by the enforcement of regulations which have been beneficial to other members of industry. Some industries, such as the capital-goods industries, have suffered from the enforcement of the codes, which has resulted in an increase in wages without an increase in the demand for their products which would make possible an increase in price.

(3) The development of the NRA program has resulted in the establishment of a bureaucratic control of business to a considerable degree. A vast amount of authority has been placed in the hands of Government employees, many of whom have not the experience to use it wisely. The possibilities for harm in this tendency are very great. The authority granted to the Government in the NIRA and in the codes would enable it to exercise a very potent influence in the management of industry in this country. No student of industrial management can view this possibility without apprehension. The extent to which this condition will be dangerous depends primarily upon the point of view and the wisdom of those who are responsible for the administration of the national recovery program. It also depends to some degree upon the extent to which industrial leaders are able to comprehend the problems with which industry is faced, and are willing and able to cooperate in developing self-government. If industry will not control itself wisely, Government control is inevitable. To avoid this, industry should be willing to make strenuous efforts to correct its ills.

SUGGESTIONS FOR FUTURE FUNCTIONING OF THE NRA

This foregoing analysis has attempted to indicate the ways in which the NRA has affected industry during the past year. As stated in the beginning, it is impossible to measure accurately the effect of the NRA on industry at this time. It is my opinion that a careful weighing of the good and the bad which has resulted from the NIRA must lead the impartial observer to conclude that to date the net effect of the act has been beneficial to industry. I sincerely believe that industry today is in a better condition than it would be if this act had not been passed. The main problem, in my opinion, is not to bemoan the past but to seek to improve the future administration of the program. I cannot refrain, therefore, from stating briefly my conception of how the NRA should function in the future.

(1) The NRA, in my opinion, should not continue to exercise legislative and judicial functions. The legislative functions which it has assumed should be exercised by Congress. I realize that at the time the original act was passed, our experience was not sufficient to enable us to define the responsibilities of the Government with reference to these problems in specific terms, but now our experience is sufficient to enable us to define the authority of the Government under the NRA program more concretely. The judicial functions which the NRA assumed should be exercised by the courts. As a consequence, the functions exercised by the NRA should be administrative only in nature and these should be divided into two major groups, namely, executive functions and staff functions. The staff functions should be the more important. The NRA should conduct research to obtain information which would enable industry to govern itself wisely, make this information available to industry, and advise and counsel with industry as to how it can become self-governing. The administrative functions of the NRA should be confined primarily to the service of its representatives as official observers in the councils of industry so that the Government may be aware of the actions of industry and may inform industry when it believes the contemplated actions are illegal or improper from the public point of view. In this case, if industry does not agree with the representatives of the NRA, the case should be referred to some agency of the Government, such as the Federal Trade Commission, which can make a decision. If industry is not satisfied with this decision, it should have the right of appeal to the courts for final adjudication.

(2) The NRA should confine its activities for a long period of time to the major industries and not seek to exercise control of the multitudinous small units of industry. It should seek to deal only with the major problems of each industry and not to remedy all the evils of industry at one time.

(3) The NRA should reorganize on a functional basis with one advisory board which would serve in a similar capacity to that of a board of directors in an industrial enterprise. Some members of this board should have term appointments and be paid for their services like other appointees of the Government. These services should be supplemented by those of members chosen from industry and labor who would continuously bring to the board a fresh point of view.

(4) And finally, the primary purpose of the activities of the Administration should be neither recovery nor reform, but rather to aid industrial leaders in the development of a sound program of economic organization for industry which will give full recognition to the rights of labor, capital, and the public.

HYDRAULIC VALVES *and* GATES *for* BOULDER DAM

Part II—The Design of the Large Butterfly Valves and Their Operating Mechanisms

By P. A. KINZIE[1]

IN THE U-shaped power plant on the downstream side of Boulder dam, Fig. 1, the ultimate installation will consist of 15 turbines of 115,000 hp and two turbines of 55,000 hp each. Seven of the large units and both small units will be installed in the Arizona leg of the "U" and the eight remaining large units in the other leg on the Nevada side of the river.

The 17 units which constitute the ultimate installation will be divided into four groups, three of which will be alike and consist of four large units each, while the fourth group will consist of three large units and two small units. Each of these four groups of turbines will be connected to one of the four 30-ft penstock headers by means of four 13-ft penstocks, each of which will lead directly to its individual 115,000-hp turbine with the exception of the last 13-ft penstock on the Arizona side of the river, the lower end of which will be bifurcated to connect with the two 55,000-hp turbines in the downstream end of the power house. By this arrangement it will be seen by inspection of the plan view of Fig. 1 that each of the four 30-ft penstock headers supplies water to one-quarter of the turbines in the power plant and, consequently, whenever one of these headers is shut down, one-fourth of the power plant will be rendered inoperative. The loss of revenue thereby incurred will be approximately $4500 per day.

In Fig. 2, which is a cross-section through the river gorge and the two wings of the power plant, the relative positions in elevation of the two upper and the two lower headers with their penstocks are shown. The butterfly valves in the penstocks adjacent to the turbines are also shown.

Conditions made it desirable to provide some means whereby the flow of water to any turbine could be shut off without closing down the other turbines fed by the same header. It was found that individual flow control to each turbine would be conducive to the general safety of the whole plant, that the overall characteristics of pressure rise in the penstock header systems could be improved by holding it within lower limits, and that the deterioration of the turbine wicket gates resulting from leakage while the turbines were idle would be avoided. For these and other reasons it was decided to provide shut-off valves for every turbine (see Fig. 2) and a survey of various types of apparatus available indicated that butterfly valves in the penstocks adjacent to each turbine would be well adapted to the operating conditions.

A survey of similar equipment previously fabricated and in use indicated that the conventional types of operating mechanisms customarily employed for opening and closing butterfly valves would, in the size of valves and with the severity of service here presented, necessitate very large and cumbersome apparatus which would occupy space in the power plant much needed for other purposes.

Various forms of mechanism were then considered for the purpose of improving this condition if possible, and after much discouraging investigation and search a solution was finally found which had many commendable features, and preliminary layouts and designs were started immediately to develop its possibilities.

In the preparation of these designs many unexpected obstacles and difficulties appeared and no sooner was one of these successfully overcome than another arose to take its place. In the work that followed much real pioneering was done, and to those who were intimately engaged in it there came, at times, not only the all too frequent sinking sensation when a seemingly insurmountable obstacle would be disclosed, but also, at rare and far less frequent intervals, the thrill of explorers adventuring into the realms of the unknown.

COMMENDABLE FEATURES OF DESIGN ADOPTED

As a result of further investigations, layouts, and estimates, the desirable features of the designs for the new butterfly valves described in this paper were found to be as follows:

(1) Great compactness with no moving parts exposed.

(2) Entirely self-contained design.

(3) Elimination of supplemental oil reservoirs.

(4) Practically no added bending moments imposed on the stems through application of turning torques in either direction, as these are applied in perfect balance.

(5) Automatic, hydraulically operated locking device, effective in any position of the butterfly leaf as soon as the oil-pump motor stops.

(6) Elimination of massive supplemental anchorages in either the floor or walls to take the reactions set up by the operating mechanism.

(7) Elimination of oil tanks and the intercommunicating piping required.

(8) All working parts easy of access, yet fully protected.

(9) Safety to visitors as well as operators assured.

(10) Attractive appearance, blending in with turbines and adjacent equipment in a most harmonious manner.

(11) Completed installation made at considerably less cost.

The compactness and freedom from exposed moving parts removed the hazard of possible interference with other power-house equipment, which could, consequently, be set in closer and more desirable, convenient, and efficient locations.

With such advantages the decision to proceed with the preparation of detail designs and purchase-specification drawings

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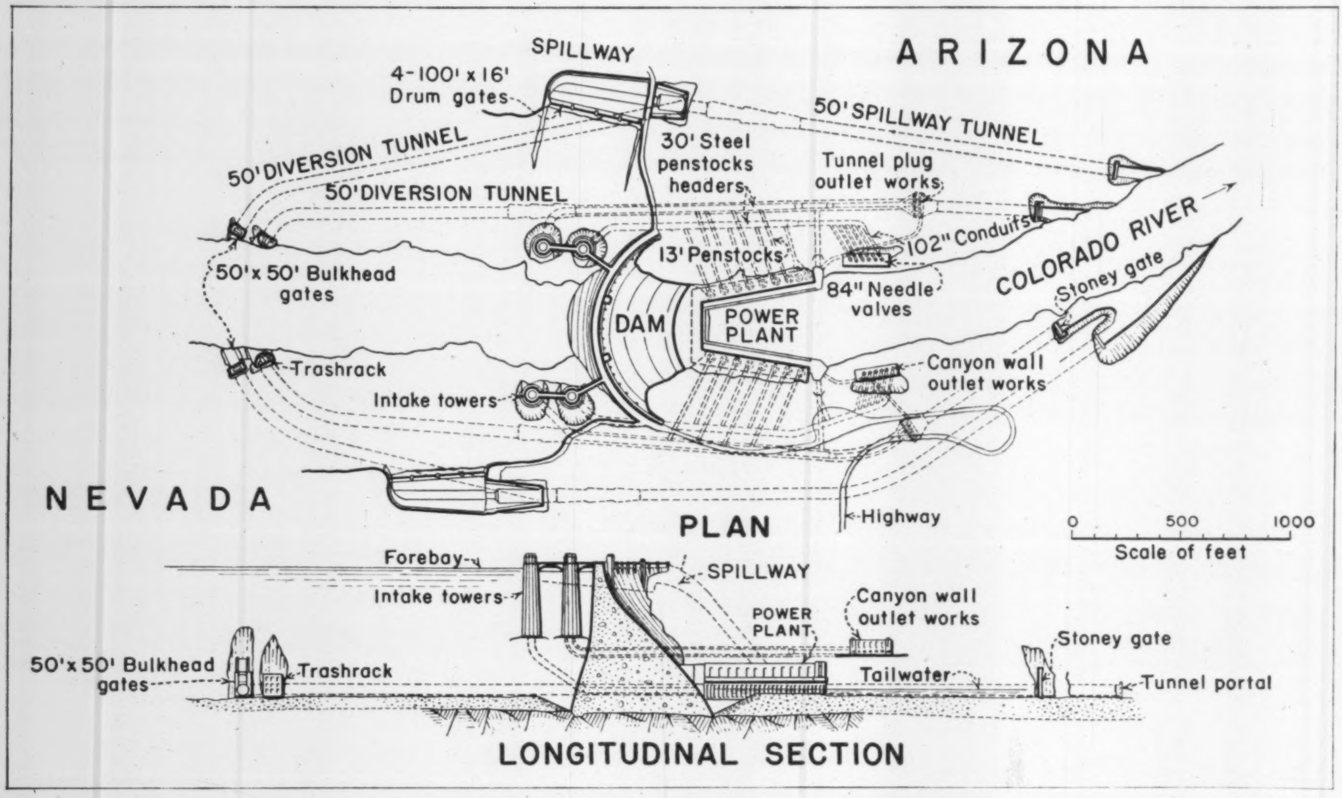

ARIZONA
SPILLWAY
4-100' x 16' Drum gates
50' SPILLWAY TUNNEL
50' DIVERSION TUNNEL
50' DIVERSION TUNNEL
30' Steel penstocks headers
Tunnel plug outlet works
102" Conduits
COLORADO RIVER
Stoney gate
13' Penstocks
50' x 50' Bulkhead gates
84" Needle valves
DAM
POWER PLANT
Trashrack
Canyon wall outlet works
Intake towers
NEVADA
PLAN
Highway
0 500 1000
Scale of feet
Forebay
SPILLWAY
Intake towers
Canyon wall outlet works
POWER PLANT
Stoney gate
50' x 50' Bulkhead gates
Trashrack
Tailwater
Tunnel portal
LONGITUDINAL SECTION

FIG. 1 PLAN AND LONGITUDINAL SECTION OF BOULDER DAM AND ASSOCIATED WORKS

was made. It was also decided to prepare designs from which the principal parts could be made of cast steel, and alternate designs for high-grade fusion-welded steel-plate construction.

DESIGN DATA

The designs were based upon the following data: Maximum design working pressure, 300 lb per sq in., maximum design test pressure, 500 lb per sq in.

(*A*) Normal operating conditions assumed a pressure of 300 lb per sq in. and a discharge through each valve of 2000 cfs.

(*B*) Emergency operating conditions assumed a pressure of 300 lb per sq in. and a discharge through each valve of 8000 cfs.

When making up the first trial designs it was decided that all of the principal stresses under condition (*A*) were to be held to within 7500 lb per sq in., and under condition (*B*) to within three-fourths of the yield strength, as it was then the intention to use medium-grade steel castings to conform with Federal specifications QQ-S-681 with a yield strength of 32,000 lb per sq in. However, when the major parts were designed to meet this requirement it was found that in many places metal as thick as 7 and 8 in. was indicated. Such thicknesses introduced hazards in the way of hidden defects which would, because of the nature and shape of these castings, be exceedingly difficult to detect either by X-ray or by the gamma-ray inspection. Core drilling was considered, but the unwieldy nature of the castings when designed to meet the conditions as first specified made it evident that, if practicable, castings should be secured with a much higher yield strength than that called for under specifications QQ-S-681.

Consultation with some of the leading steel-foundry authorities revealed that with but slight increase in price, the major parts, such as valve bodies, leaves, rotor cylinders, etc., could be made as so-called "alloy-steel castings," possessing the following composition and physical properties: Carbon, 0.25 to 0.35 per cent; manganese, 1.00 per cent, maximum; silicon, 0.4 per cent, maximum; nickel to be added if necessary; ultimate tensile strength, 75,000 lb per sq in.; yield point, 50,000 lb per sq in.; elongation in 2 in., 25 per cent; and reduction in area, 50 per cent.

Having established the availability of castings possessing these characteristics, the design data were revised as follows: All principal stresses under condition (*A*) were to be held to within one-third, and under condition (*B*) to within three-fourths of the yield strength of the materials employed.

The minimum closing time for the valves was set at 4 min or longer. Careful pressure-rise studies of each of the four sets of complex penstock headers with their manifolds, quadruple penstock branches, and sextuple needle-valve outlets clearly indicated the necessity of avoiding rapid changes of velocity of flow within any of the four systems because of the reverberatory or reflected pressure wave which would become active were a shorter closing time permitted.

For each valve, two 12-in. by-pass pipes with 12-in. cooling-water connections were required with electrically operated needle valves for service use and manually operated gate valves to protect the needle valves.

OPERATING CONDITIONS

The butterfly valves described in this section are to be used as shut-off valves and are to be installed in the power house at the lower end of each 13-ft penstock, with a tapered connecting pipe extending from the downstream flange of each valve to the inlet flange of the turbine scroll case. (See Fig. 3.) Under normal conditions these valves will be operated with no flow through the penstock and valve other than that caused by leakage through the turbine wicket gates. They will not be closed until after the wicket gates are closed, and they will not be opened until after the 12-in. by-pass needle valves have opened, the turbine scroll cases have been filled, and the pressures therein built up to substantially the same as those in the penstocks. The valves will be operated under a maximum head, with water hammer included, of 625 ft, but are capable of safe closure within a period of 5 min under a pressure of 300 lb per sq in. and a flow of 8000 cfs.

In Fig. 3 a vertical transverse section through one of the power-house wings on the centerline of one of the large units, shows the relative arrangement of the 82,500-kva generator, the 115,000-hp turbine, the connector pipe extending from the turbine scroll case to the left to the butterfly valve, and the 13-ft penstock extending to the left from the butterfly valve. One of the two 300-ton cranes provided in each wing of the power plant is shown above the generator immediately beneath the roof trusses. These cranes will reach down through hatches in the floor to handle the butterfly valves.

Each valve is supported upon bifurcated reinforced-concrete piers capped with semi-steel sole plates, finished upon their upper surfaces to receive the finished base pads of the valve bodies and anchored to the pier tops by heavy bolts. The valve bodies are clamped to the sole plates by short bolts whose removal frees the valves from their foundation anchorage, permitting them to be slid downstream, with the upper sole-plate faces acting as skid rails, away from the penstock connection, after the pipes to the turbine casings have been removed. A special sister hook provided for that purpose can be screwed into the end of the upper rotor stem (see longitudinal section of Fig. 4) and the cranes can lift the complete valve with its hydraulic rotor out of the penstock line or set it into place therein, as the case may be.

DESCRIPTION OF THE BUTTERFLY VALVE

In Fig. 4 the complete assembly of the butterfly valve and its hydraulic rotor is shown in vertical section, and in downstream elevation looking upstream through the wide-open valve.

The connecting pipes between the downstream flanges of the butterfly valves and the inlet flanges of the turbine casings are cut to lengths after the turbines have been embedded in concrete and the butterfly valves bolted to their penstock flanges. The upstream ends of the connector pipes telescope into bored flanges bolted to the downstream flanges of the butterfly valves. The downstream ends of these connector pipes are riveted into flanges which are bolted to the mating flanges of the turbine scroll cases. The upstream end of each connector pipe is cut to fit, telescoped into its upstream flange, and lowered to place, and the flanges are bolted. Then, using the rivet holes in the ferrule portion of the upstream flange as a drilling jig, these rivet holes are drilled in the connector pipe. By this arrangement any inaccuracies likely to occur in the setting can be taken care of after the turbines and valves are in their final positions.

The static load on the valve leaf as a result of the pressure of 300 lb per sq in. is 6,500,000 lb, and under a pressure of 500 lb per sq in. the load is 10,800,000 lb. These static loadings on the leaf produce stem-bearing loadings from the upper and lower stems of 3,250,000 and 5,440,000 lb (see Fig. 5), respectively. To carry these concentrated loadings into the valve bodies and distribute them with sufficient

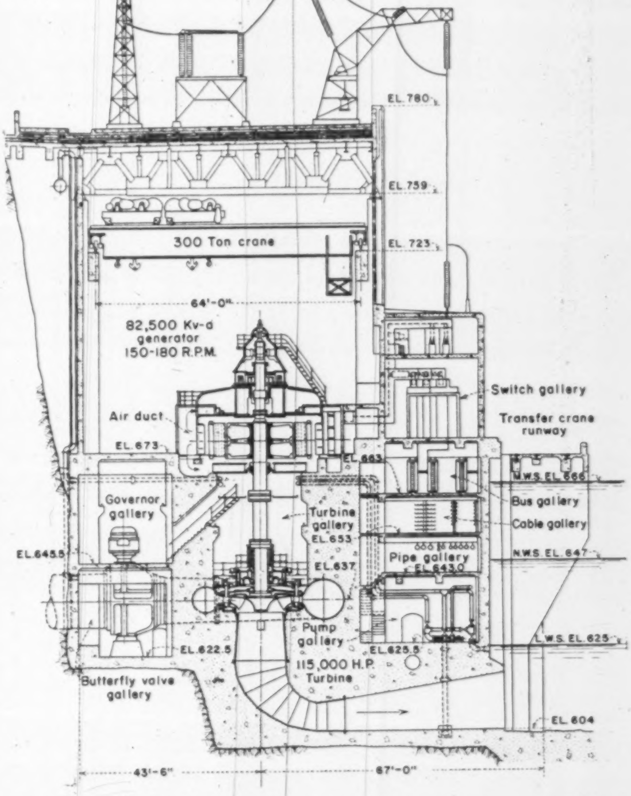

EL. 780
EL. 759
EL. 723
300 Ton crane
64'-0"
82,500 Kv-a generator 150-180 R.P.M.
Switch gallery
Transfer crane runway
Air duct
EL. 673
EL. 663
N.W.S. EL. 666
Governor gallery
Turbine gallery EL. 653
Bus gallery
Cable gallery
EL. 645.5
EL. 637
Pipe gallery EL. 643.0
N.W.S. EL. 647
Pump gallery
EL. 622.5
EL. 625.5
L.W.S. EL. 625
115,000 H.P. Turbine
Butterfly valve gallery
EL. 604
43'-6"
67'-0"

FIG. 3 CROSS-SECTION THROUGH ONE OF THE POWER-PLANT WINGS

(The relative arrangement of the butterfly valve and its connection to the turbine scroll case is shown.)

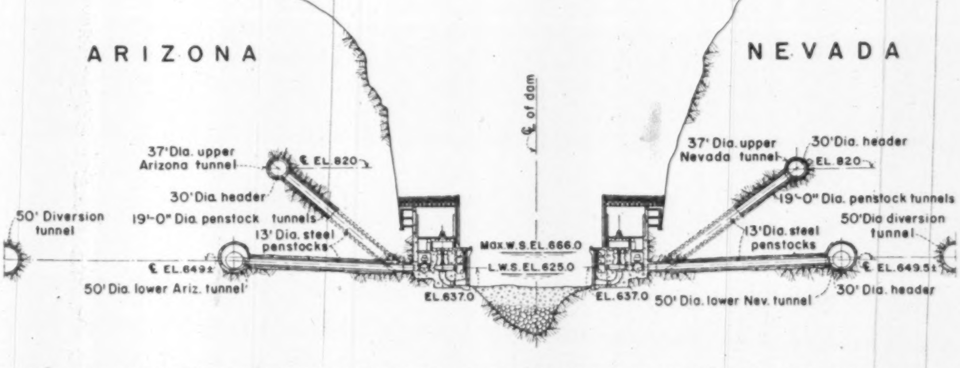

TOP OF DAM-ELEVATION 1232
Max. W. S. El. 1221.0
ARIZONA
NEVADA
℄ of dam
37' Dia. upper Arizona tunnel
℄ EL. 820
30' Dia. header
50' Diversion tunnel
19'-0" Dia. penstock tunnels
13' Dia. steel penstocks
℄ EL. 649±
50' Dia. lower Ariz. tunnel
EL. 637.0
Max. W. S. EL. 666.0
L. W. S. EL. 625.0
EL. 637.0
37' Dia. upper Nevada tunnel
30' Dia. header
EL. 820
19'-0" Dia. penstock tunnels
50' Dia diversion tunnel
13' Dia. steel penstocks
℄ EL. 649.5±
50' Dia. lower Nev. tunnel
30' Dia. header

FIG. 2 CROSS-SECTION THROUGH THE TWIN WINGS OF THE POWER PLANT AND THE ADJACENT CANYON WALLS, SHOWING THE RELATIVE LOCATIONS OF THE UPPER AND THE LOW-LEVEL PENSTOCK HEADERS

(The butterfly valves are in the penstocks adjacent to the turbines.)

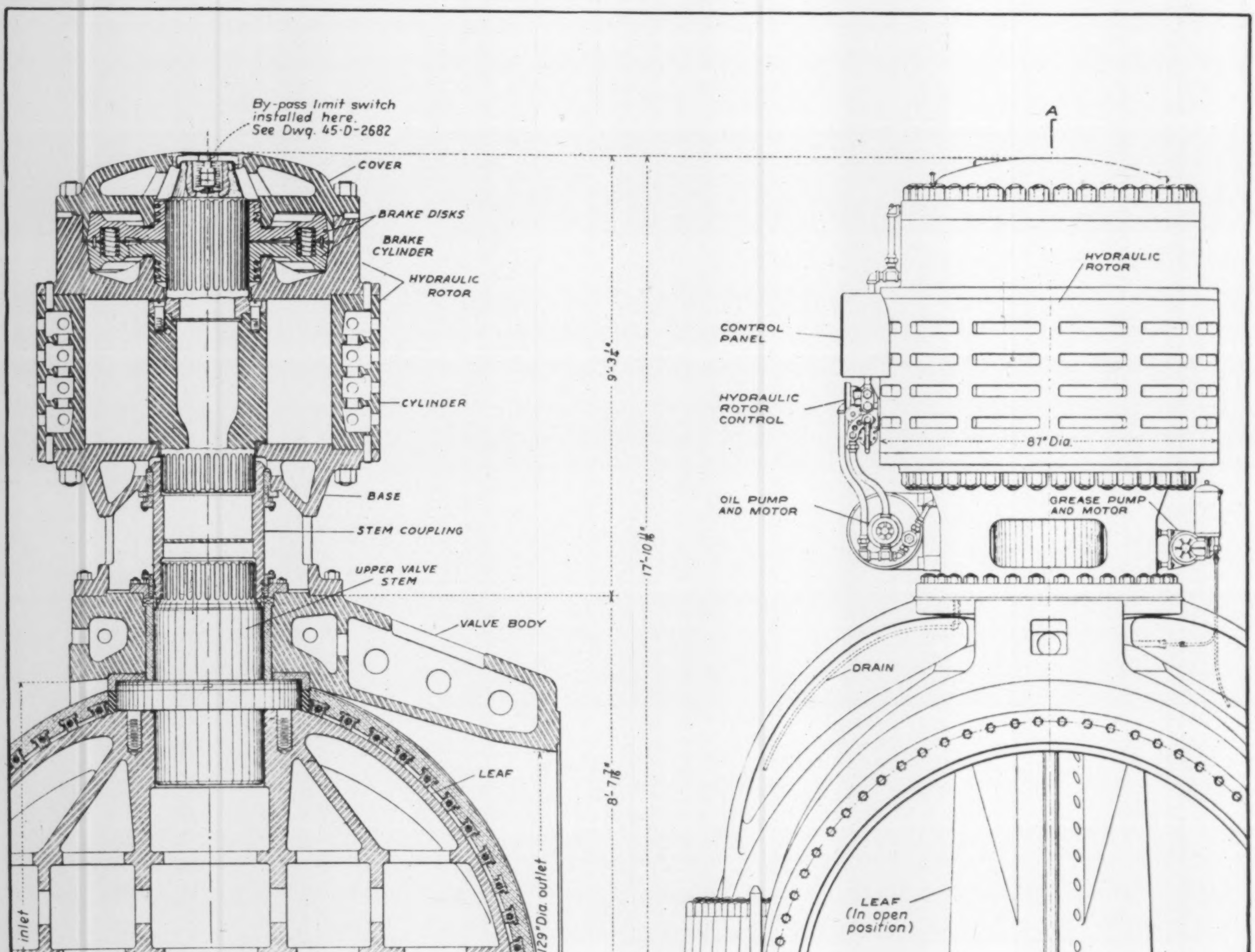

By-pass limit switch installed here. See Dwg. 45-D-2682
COVER
BRAKE DISKS
BRAKE CYLINDER
HYDRAULIC ROTOR
CYLINDER
BASE
STEM COUPLING
UPPER VALVE STEM
VALVE BODY
LEAF
FLOW
165⅝" Dia. inlet
129" Dia. outlet
VALVE BODY
3'-0"
7'-6"
9'-3½"
17'-10 11/16"
8'-7⅞"
10'-5⅞"
8'-7⅞"
22"
VERTICAL SECTION A-A
WITH VALVE OPEN
A
HYDRAULIC ROTOR
CONTROL PANEL
HYDRAULIC ROTOR CONTROL
87" Dia.
OIL PUMP AND MOTOR
GREASE PUMP AND MOTOR
DRAIN
LEAF (In open position)
Horiz. ℄ of valve - El. 637.0
Symm. about ℄ except as shown
4'-6"
15'-11⅛"
EQUALIZED PRESSURE SWITCH
12" BY-PASS NO. 1
4'-1"
4'-1"
A
DOWNSTREAM ELEVATION
VALVE AND ROTOR
ROTOR CONTROL FOR NEVADA VALVES SHOWN
ROTOR CONTROL FOR ARIZONA VALVES OPP. HAND

FIG. 4 COMPLETELY ASSEMBLED BUTTERFLY VALVE, IN THE OPEN POSITION, AND HYDRAULIC ROTOR

uniformity around the circumference of the upstream body flanges and to transfer them to the penstock flanges without undue concentration of load upon the bolts closest to the stem bushings, without inviting progressive failure of these bolts, was a problem. In the final solution they were carried fanwise to the downstream flanges of the valve bodies by means of heavy box-girder members cast integrally with the upper and lower valve bodies and aligned with the vertical axes of the valves. (See Figs. 4, 5, and 6.) By so doing the stresses were distributed around the circumference of the body shell by the time they reached back to the upstream flanges of the valves. A heavy circumferential ribbing aligned with the stem hubs and normal to the flow through the valves was required to take the vertical reactions produced by the downstream thrusts from the stems upon the box girders. (See Fig. 6.)

It was necessary to carry the static water loads to the upstream valve flanges because these valves will, in certain phases of plant construction, be installed and connected to their respective penstocks, their leaves closed, and then subjected to full reservoir pressure for some time prior to the installation of the turbines to which they will eventually be connected.

STEM BUSHINGS

The heavily loaded stem bushings presented another serious problem with respect to uniform distribution of bearing pressures within them, for calculations showed that the combined flexure of the leaf and the stems acting as simple beams under load would cause

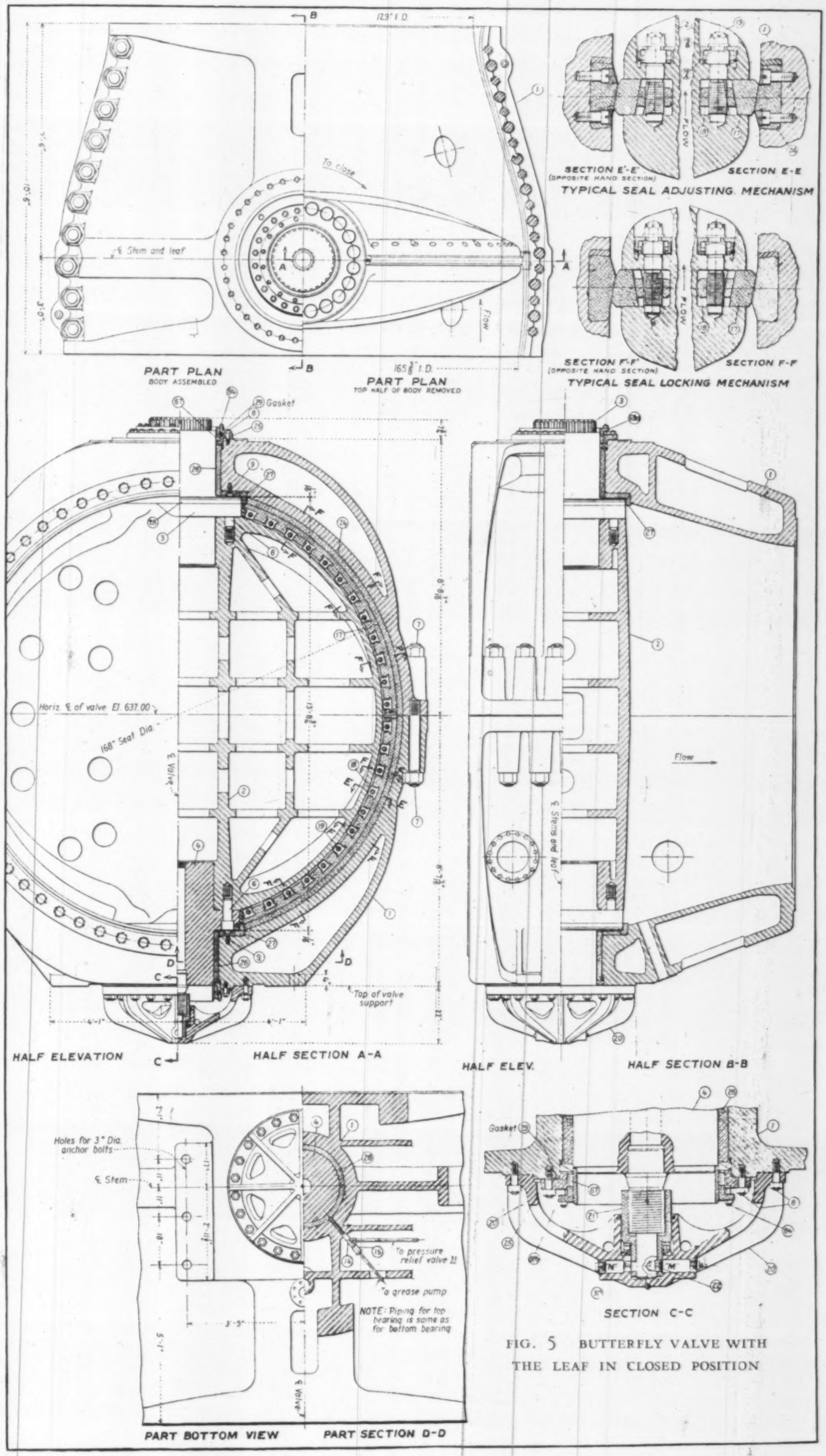

123" I.D.
To close
℄ Stem and leaf
Flow
165⅝" I.D.
PART PLAN
BODY ASSEMBLED
PART PLAN
TOP HALF OF BODY REMOVED
SECTION E'-E'
(OPPOSITE HAND SECTION)
SECTION E-E
TYPICAL SEAL ADJUSTING MECHANISM
SECTION F'-F'
(OPPOSITE HAND SECTION)
SECTION F-F
TYPICAL SEAL LOCKING MECHANISM
Gasket
Horiz. ℄ of valve El. 637.00
168" Seat Dia.
℄ Valve
Top of valve support
HALF ELEVATION
HALF SECTION A-A
HALF ELEV.
HALF SECTION B-B
℄ Stems and leaf
Holes for 3" Dia. anchor bolts
℄ Stem
To pressure relief valve
To grease pump
NOTE: Piping for top bearing is same as for bottom bearing
PART BOTTOM VIEW
PART SECTION D-D
SECTION C-C

FIG. 5 BUTTERFLY VALVE WITH THE LEAF IN CLOSED POSITION

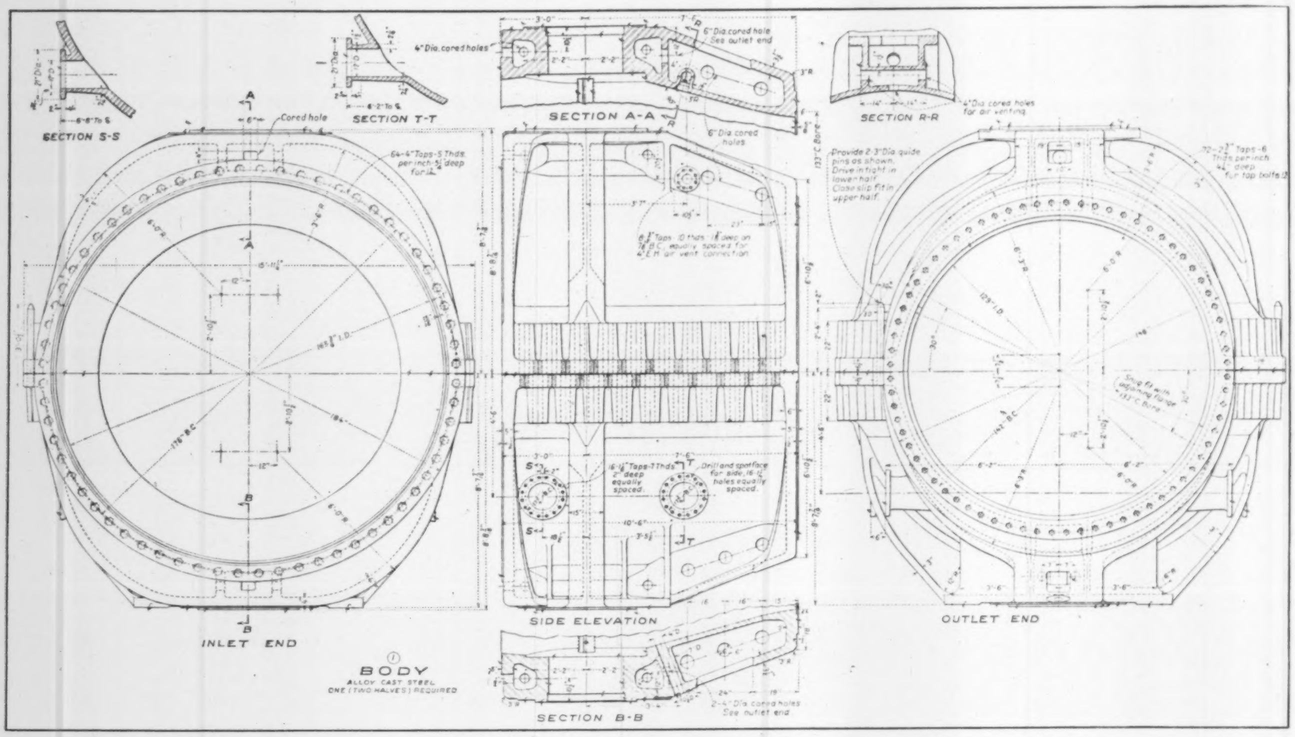


FIG. 6 DETAIL CONSTRUCTION OF THE UPPER AND LOWER VALVE-BODY CASTINGS

(The heavy box ribbing from the stem hubs to the outlet flange and the circumferential ribbing from these hubs to the longitudinal body joint, for distributing the heavy stem loadings into the body walls to secure uniform loading on the inlet-flange bolts, are shown in these views.)

the stems to throw their loads on to the end surfaces of the bushings adjacent to the leaf, and this concentration of pressure would be sufficient to crush the metal of the bushings.

Bushings with turned, spherical, external surfaces fitting into spherical surfaces in the valve bodies would permit the bushings to align themselves with the stems as they were deflected, and thus provide practically uniform bearing distribution over the entire projected area of the bushing surfaces. However, such ball-and-socket construction for use with stems 28 in. in diameter was obviously impracticable.

As a precautionary measure, designs and layouts of this ball-and-socket type of bushing mounting were prepared, but they resulted in gigantic proportions. It was finally demonstrated that such a solution was prohibitive from an economic standpoint and would not be satisfactory on account of the added joints involved in the valve-body castings.

Other possibilities were explored only to reveal that they also possessed some major defect. Finally, when the outlook had become most discouraging, a workable solution was found. In reviewing what had been considered in an effort to solve this problem, and the governing conditions involved, it was established that the angular deflection of the stems under maximum conditions of loading was relatively small, that it was most desirable that the bushings should move in unison with the stems as they deviated from their unloaded positions of axial alignment to their fully loaded positions, and that it was desirable to move the points of support closer together as the loadings were increased in order to decrease the bending moments. These considerations suggested that the bushings be allowed to rock in synchronism with the distortions occurring in the leaf and the stub stems attached to its opposite ends as deflection increased, in essentially the same manner that a simple beam would deflect under similar conditions.

Further study disclosed the fact that if the exterior cylindrical surfaces of the bushings were to be given a very slight curvature (see Fig. 7, section *F-F*) so that they could rock on the downstream surfaces of the valve-body hubs, they would then automatically accommodate themselves to the beam flexures occurring within the butterfly leaf and its stub stems and would likewise permit an accompanying reverse movement of the hubs in the valve body produced by the action of the box-girder members (see Figs. 4, 5, and 6). This would tend to force the hubs to deflect so that their centerlines would move in a direction opposite to that simultaneously occurring in the leaf and stems. The rocking action of the bushings is illustrated in Fig. 8.

Several problems are involved in the design of a rocking bushing such as that employed in these butterfly valves. First, the maximum angular deflection of the shaft or stem and its cooperating parts must be determined. Second, the shape of the rocking surface, which can be either of constant radius or of parabolic curve, must be chosen. Third, inasmuch as the curved surface of the bushing is not spherical, clearance must be provided between the central major diameter and the cylindrical bore of the hub within which it is mounted in order to eliminate binding and interference. Fourth, the point of contact with respect to the length of the bushing when under full load must be far enough back from the end of the bushing to permit adequate bearing area. Fifth, the bushing, while being left relatively free to follow the deviations from alignment of the stem under varying loadings, must be retained in the same axial position with respect to the length of the hub bore.

For convenience in calculation, a parabolic curve was used. The equation of this parabola and the central-diameter clearance can be determined as soon as the location of the center of the shaped contact area between the bushing and the hub bore for the fully loaded position has been determined.

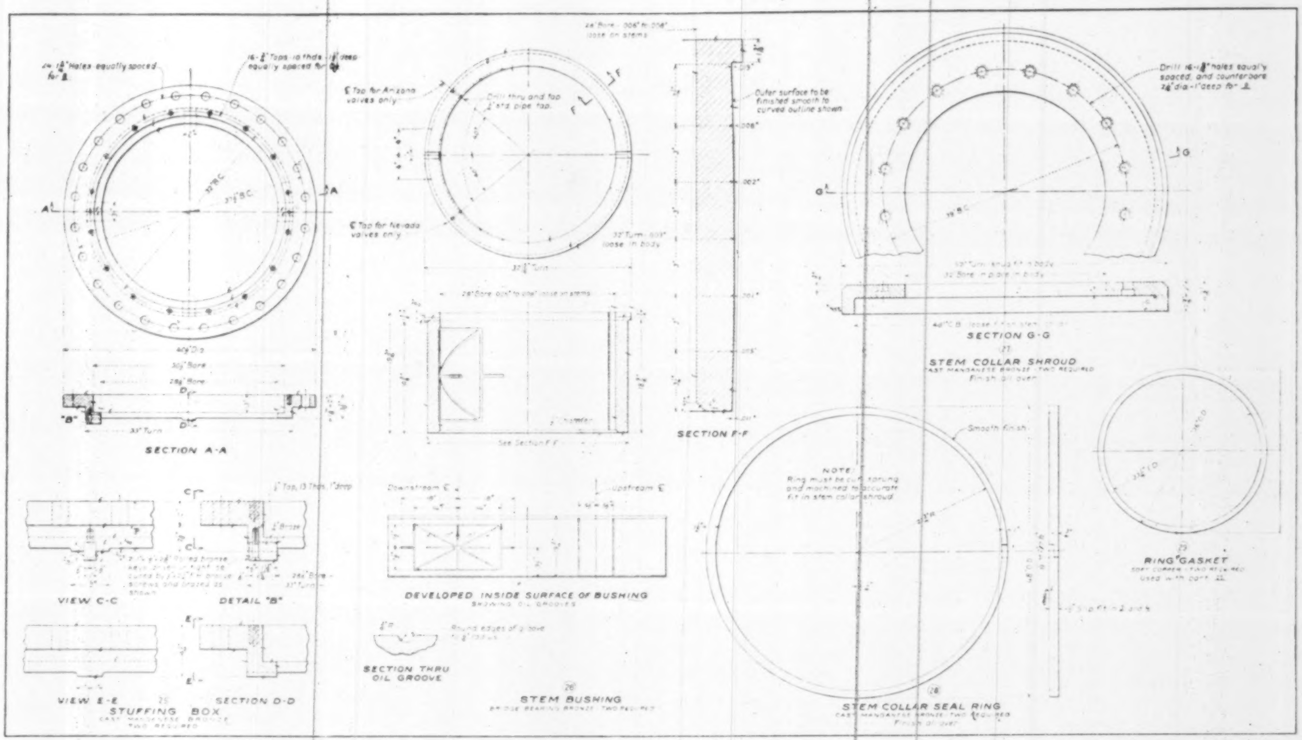


FIG. 7 DETAILS OF CONSTRUCTION OF THE 28-IN. ROCKING BUSHINGS IN WHICH THE GATE STEMS ARE CARRIED

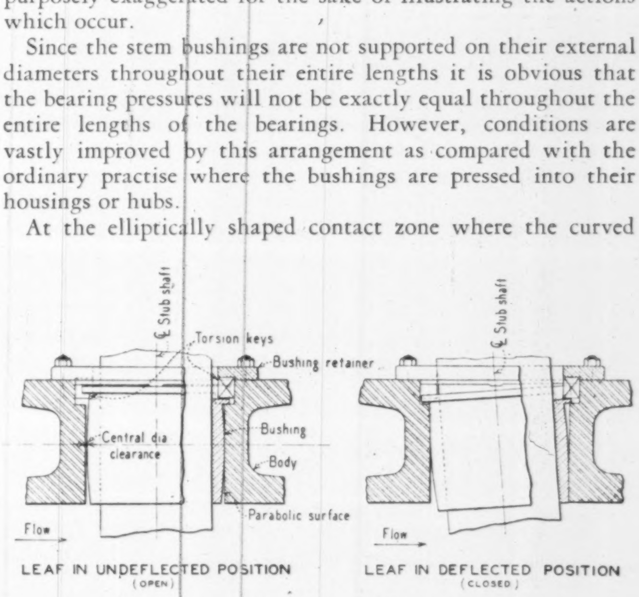
The problem of maintaining the bushing in its correct position lengthwise with respect to the hub was the most difficult of all, and for a time threatened the successful use of this device. It was found that there was a bearing load in excess of 5,000,000 lb on each bushing and that even with the most efficient form of lubrication the bushing would slide upon the stem as flexure and contraflexure occurred. This would result in a tendency to move the bushing lengthwise, and the force so generated was so prodigious as to make any retaining collar or similar restraining mechanism inadequate to resist it.

In the solution finally developed a small raised block was placed on the under side of the bushing lip on its downstream side where practically no endwise movement with respect to the hub occurred, thereby restraining the bushing from dropping downward, and two raised blocks were placed opposite to each other on the under face of the seat retainer, with their common centerline at right angles to the centerline of the single block on the lip of the bushing.

The left-hand view of Fig. 8 shows the lip on the under side of the stem collar which prevents the bushing from moving downward, and one of the two raised blocks on the under side of the bushing retainer which prevent the bushing from moving upward. In the right-hand view the lip on the under side of the stem collar has slid somewhat to the left but has not moved vertically to any appreciable extent. It will be seen that the bushing has rolled on its curved surface and that the contact has now moved downward. Further study of these views will show that the supporting lip and raised blocks have in no way interfered with the rocking motion of the bushing, while at the same time they are fully effective in maintaining the bushing in its correct position endwise with respect to the hub. Torsional keys in the under face of the bushing retainer prevent the bushing from rotating but offer little resistance to the rocking movement of the bushing as they are aligned with the centerline of flow through the valve. The relative displacements and movements as depicted in the views of Fig. 8 are purposely exaggerated for the sake of illustrating the actions which occur.

Since the stem bushings are not supported on their external diameters throughout their entire lengths it is obvious that the bearing pressures will not be exactly equal throughout the entire lengths of the bearings. However, conditions are vastly improved by this arrangement as compared with the ordinary practise where the bushings are pressed into their housings or hubs.

At the elliptically shaped contact zone where the curved surface of the bushing is in contact with the hub wall there is rolling action but no rubbing occurs, and it has been found that the contact pressures in this zone are quite moderate, being less than 15,500 lb per sq in.



LEAF IN UNDEFLECTED POSITION (OPEN) LEAF IN DEFLECTED POSITION (CLOSED)

FIG. 8 ROCKING ACTION OF THE STEM BUSHINGS AND THE CHANGE OF POSITION FROM THE UNLOADED CONDITION (LEFT) TO THE MAXIMUM TEST LOAD (RIGHT)

Another difficulty closely related to these rocking bushings was to provide some means whereby the freedom of angular displacements and deflections of the upper stems would be left unimpaired, and yet permit the transfer of torques of high magnitude generated by means that will be described later, from these mechanisms to and into the stems.

One solution for this problem was worked out satisfactorily by continuing the upper stem upward into the hydraulic rotor, mounting the rotor directly upon the stem, and then floating the base flange of the hydraulic rotor mechanism between two heavy rings of cushion-stock rubber, so that the entire rotor mechanism could weave and gyrate in unison with the upper gate-stem extension.

With this form of construction the length of the upper stems were more than 13 ft and there arose the question of the practicability of machining these parts, particularly the splines for the rotor and brake disks. Shipping difficulties were also anticipated as the shaft would have to be loaded and unloaded while attached to its gate leaf, whose weight alone is in excess of 90,000 lb. Moreover, it was found to be desirable to assemble and test each hydraulic-rotor mechanism before placing it on its butterfly valve, and this could not be done if the upper stem was made in one piece.

These considerations made it evident that the upper butterfly-valve stem should terminate a short distance above the upper extremity of its stem bushing and that the hydraulic-rotor mechanism should be provided with its own stem member whose lower extremity would be above the plane of its bottom flange and permit the rotor unit to be lifted from the valve and set down on its own flange. This indicated some form of flexible coupling capable of transmitting the maximum runaway torque and at the same time providing the flexibility needed for the upper rocking stem bushing.

Calculations showed that normal operating conditions for design as defined in (*A*) would require a turning torque of about 26,000,000 in-lb, while under emergency operating conditions (*B*) the dynamic action of high-velocity flow acting upon the leaf as it started to close would produce a peak closing torque of approximately 48,000,000 in-lb.

It was first thought that the typical cruciform coupling construction, almost universally used in the steel-mill rolls, could be adapted in this case, but detailed investigation revealed that such a construction was not adequate. Many other devices were examined and discarded and finally of two means which would satisfactorily meet the conditions, a tubular sleeve with longitudinal splines at each end to engage similar splines machined in the opposing stems, shown in Figs. 4 and 9, was chosen.

HYDRAULIC ROTOR

By reference to the two views of Fig. 4 it will be seen that the hydraulic-rotor base is bolted fast to a flange on the upper half of the butterfly valve body, that it is aligned vertically with the centerline of the butterfly valve stems, and that it is flexibly coupled to the upper end of the upper valve stem by means of the splined sleeve.

Referring to vertical section *A-A* of Fig. 4, it will be seen that the major parts of the hydraulic rotor are (1) the base, whose lower flange is bolted to the valve body; (2) the cylinder, which is bolted to the flange of the base; (3) the brake cylinder, which is bolted to the upper face of the cylinder; and (4) the cover, which is bolted to the upper flange face of the brake cylinder. All of these parts are aligned with the vertical axis of the valve stems when in their unloaded condition. Inside the cylinder is the rotor which imparts the turning effort through the flexible coupling to operate the leaf. The upper extension of the rotor is splined to receive the brake disks in the brake cylinder above. Fig. 9 is a vertical section through the hydraulic rotor in which the detail construction of the parts can be seen.

Fig. 10, section *C-C*, is a horizontal section taken through the hydraulic rotor about midway between the lower and upper flanges of the cylinder and shows the interior walls of the cylinder with the two stator elements with vertical rectangular seal bars in their noses resting against the cylindrical hub of the rotor whose vanes carry similar bars which bear against the cylinder walls. The stators and the rotor vanes divide the interior of the rotor cylinder into four pressure chambers.

In detail *F* of Fig. 10 the rectangular seals in the stator noses are shown with the double check valves whereby high-pressure oil from whichever side of the stator it is present is admitted into the space behind the bars to force them into contact with the rotor surface. In detail *E*, the rectangular seals in the ends of the rotor vanes are shown with their double check valves to admit high-pressure oil to insure fluid-tight contact between the seal bars and the cylinder walls.

Section *G-G* is a vertical section of the stator seal mechanism with the retaining screws, the cup-leather packings behind the seals, and the coil springs which keep these seals in contact when the oil pressure has dropped to zero, while *H-H* is a vertical section through the rotor seals and shows similar details of the vertical seals in the ends of the vanes and likewise those for the upper and lower horizontal seals between the face of the rotor and the adjacent face of the brake cylinder.

Seal rings are inserted in annular grooves of rectangular cross-section cut into the under face of the brake cylinder and the upper face of the rotor base. The outer diameters of these two rings are the same as that of the rotor hub, against whose end faces these seal rings are forced by the oil pressure in the same manner as that employed for maintaining contact in the rotor and stator seals. Vertical sections through these rings may be seen in Fig. 10, sections *G-G* and *H-H*, where it will be seen that they are likewise provided with the same type of retaining screws, cup-leathers, and coil springs for forcing them out into contact when there is no oil pressure in the hydraulic chambers.

OIL PRESSURE SYSTEM

Referring again to section *C-C* of Fig. 10, it will be seen that two tubes, whose relative positions in plan resemble the letter *X*, pass through the hub of the rotor so that the upper left-hand pressure chamber is in communication with the lower right-hand pressure chamber; and, similarly, there is communication between the lower left-hand and the upper right-hand chambers.

Examination of the left-hand stator in section *C-C* will disclose four tubes cast into the metal of the rotor cylinder and stator. All four of these tubes extend to the left to the hydraulic-rotor control, while the two central tubes continue to the right to communicate with the upper left-hand pressure chamber above the stator, and to the lower left-hand pressure chamber below the stator. The two outer tubes above and below those just described both communicate with the interior of the stator, whose upper extremity is in communication through cored passages with the oil-storage cavity in the cover (see Figs. 4 and 9). The lower extremity of this cored space in the stator is in communication with the oil-storage cavity in the upper portion of the rotor base shown in Figs. 4 and 9.

When the hydraulic-rotor control at the left center of section *C-C* of Fig. 10 is set to deliver high-pressure oil to the upper left pressure chamber, pressure will be communicated through the tube in the rotor hub into the lower right pressure chamber and the high-pressure oil acting in these two chambers against

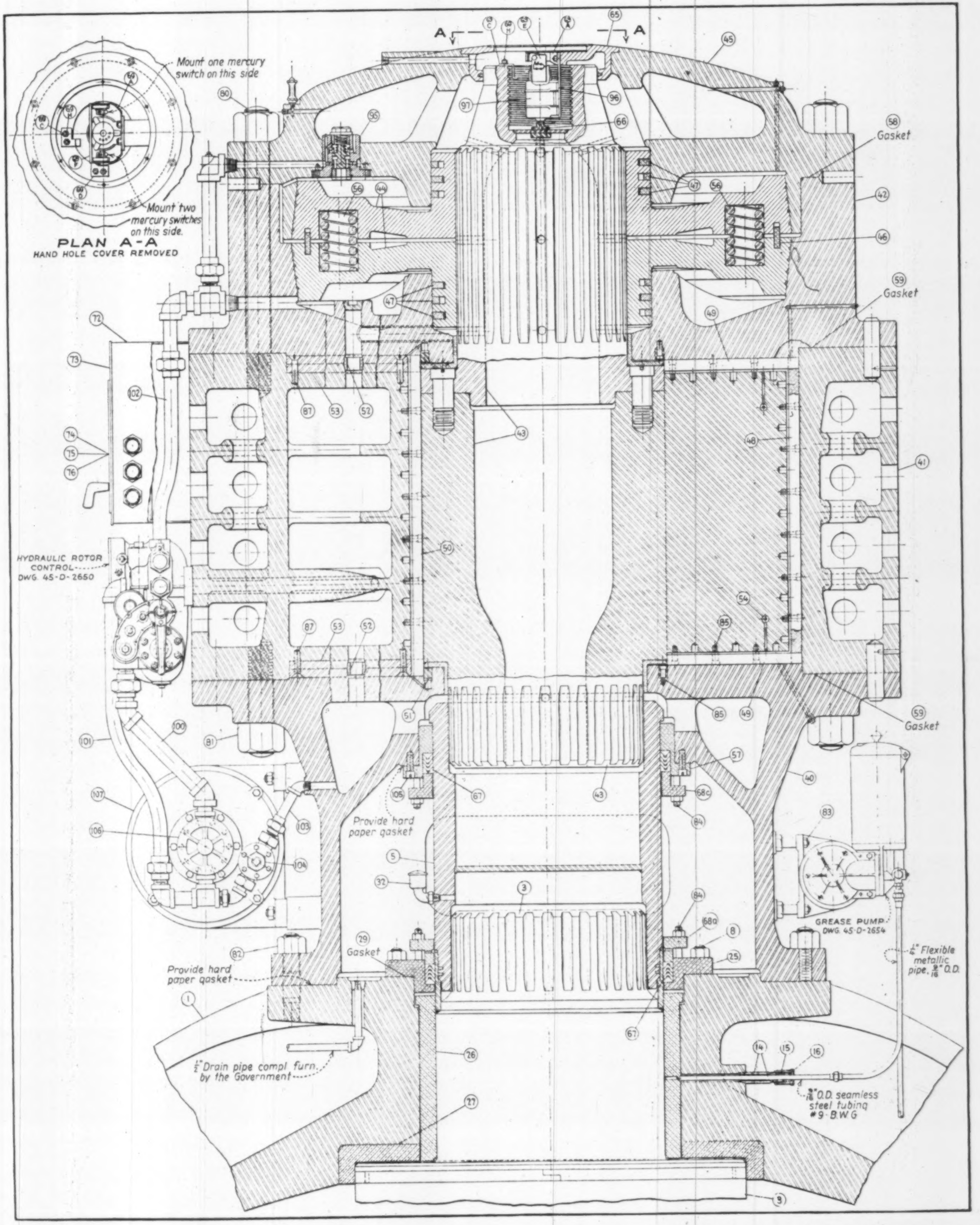

A
A
65
45
Mount one mercury switch on this side
Mount two mercury switches on this side.
PLAN A-A
HAND HOLE COVER REMOVED
97
96
66
95
80
58
Gasket
42
56
44
47
46
59
Gasket
49
72
73
102
87
53
52
43
48
41
74
75
76
HYDRAULIC ROTOR CONTROL DWG. 45-D-2650
50
54
85
51
81
101
100
59
Gasket
57
40
105
67
68c
84
107
103
83
106
Provide hard paper gasket
5
32
3
104
68a
8
GREASE PUMP DWG. 45-D-2654
82
29
Gasket
25
Provide hard paper gasket
$\frac{1}{4}$" Flexible metallic pipe, $\frac{9}{16}$" O.D.
1
14
15
16
$\frac{1}{2}$" Drain pipe compl. furn. by the Government
26
$\frac{9}{16}$" O.D. seamless steel tubing #9-B.W.G
77
9

FIG. 9 VERTICAL LONGITUDINAL SECTION THROUGH THE COMPLETELY ASSEMBLED HYDRAULIC ROTOR, FLEXIBLE COUPLING, AND AUTOMATIC HYDRAULIC BRAKING DEVICE

(The rotor-control mechanism appears on the left-hand side beneath the control panel, with the oil pump and motor mounted on the base below. The high-pressure motor-driven grease pump is shown on the opposite side of the base.)

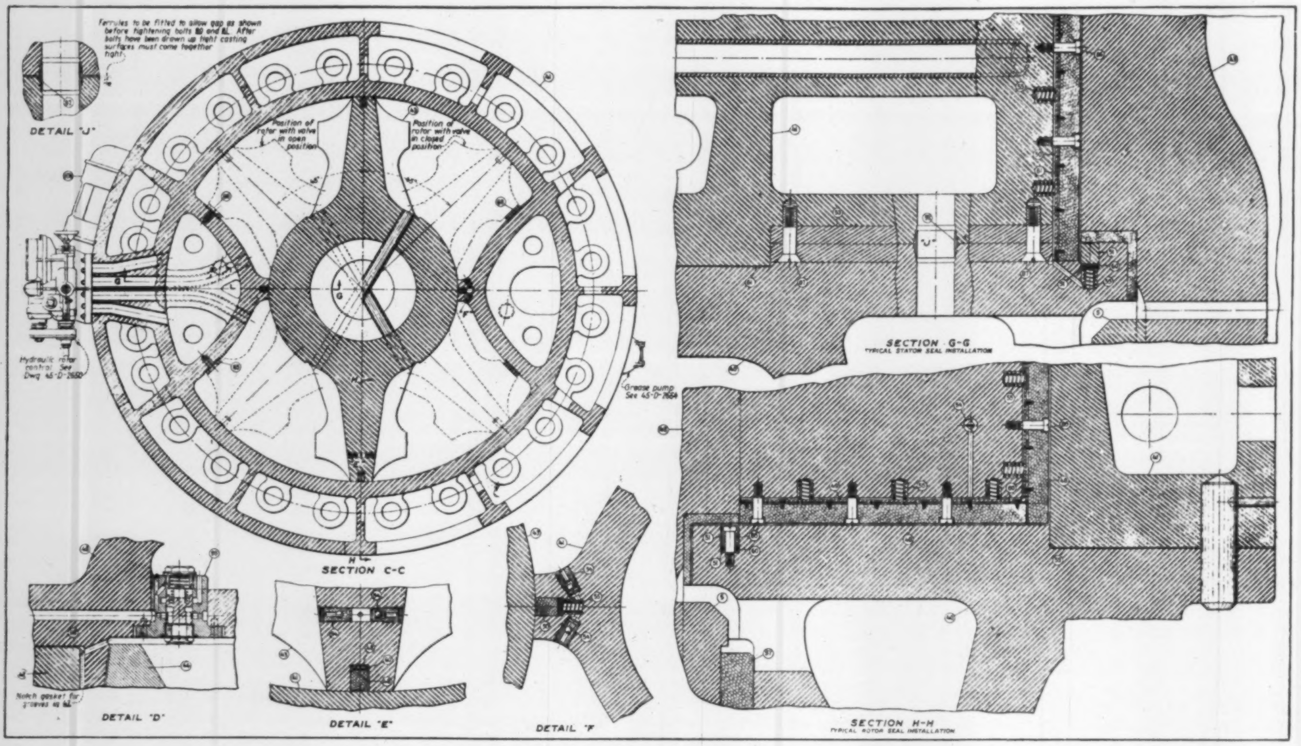

Ferrules to be fitted to allow gap as shown before tightening bolts 80 and 8L. After bolts have been drawn up tight casting surfaces must come together tight.
DETAIL "J"
Position of rotor with valve in open position
Position of rotor with valve in closed position
Hydraulic rotor control See Dwg 45-D-2650
Grease pump See 45-D-2664
SECTION C-C
"J"
SECTION G-G
TYPICAL STATOR SEAL INSTALLATION
Notch gasket for grooves at 8L
DETAIL "D"
DETAIL "E"
DETAIL "F"
SECTION H-H
TYPICAL ROTOR SEAL INSTALLATION

FIG. 10 HORIZONTAL SECTION THROUGH THE ROTOR CYLINDER WITH THE ROTOR MIDWAY BETWEEN THE EXTREMES OF ITS TRAVEL

the sides of the rotor vanes will cause clockwise rotation of the rotor and close the butterfly valve below. As this closing movement occurs, the oil in the inactive chambers is expelled from the upper right-hand chamber through the tube in the rotor hub into the lower left-hand chamber and from there through the lower of the two central tubes in the stator to and through the hydraulic-rotor control to the suction side of the rotary pump below.

The oil pump is mounted directly upon the end bell of the motor which drives it, and the motor, in turn, has its footings bolted to the side of the rotor base. The general relationship of these parts and of the hydraulic-rotor control with respect to the control panel may be seen in elevation in Fig. 11, which shows an exterior view of the hydraulic rotor from the control station.

In this view two dials will be seen near the top of the control panel. The left-hand dial is selsyn-motor driven from the sending selsyn in the top of the rotor stem (see Fig. 9), and this dial indicates the position of the butterfly-valve leaf in the body below, which has a maximum rotative travel of 90 deg. The right-hand dial indicates the oil pressure in the active chambers of the hydraulic rotor whenever it is in operation.

In the central panel below and between the two dials just mentioned are four small bull's-eye jeweled lights arranged horizontally in pairs of green and red. These are the signal lights from the two electrically operated 12-in. by-pass needle valves in the dual by-pass lines on the butterfly-valve body below to indicate when these valves are open or closed. In the central bottom portion of this center panel is the small handle which swings through an arc of about 90 deg to the right to open the butterfly valve and to the left to close it.

A knob in the left-hand panel is turned clockwise to cut out the remote-control circuits from the central power-house control pulpit from which complete automatic remote control with indication is normally affected whenever direct control from this panel on the hydraulic rotor is desired. The panel on the right gives access to the pressure-limit switch which shuts down the oil-pump motor whenever the oil pressure rises above its upper safety limits.

Behind the control panel is a mounting panel on which are fastened all the automatic electric interlocks, relays, and other equipment essential to the correct functioning and safe operation of this mechanism, and which govern the fractional-horsepower motors actuating the by-pass needle-valve controls and the hydraulic-rotor control.

Butterfly-valve-position limit switches for shutting down the oil-pump motor whenever the valve leaf reaches the fully opened or closed positions are provided in the top cover of the hydraulic rotor (see Fig. 9), as shown in larger scale in plan *A-A*. Another mercury switch is installed here, which prevents the turbine wickets from being opened until the butterfly valve has opened a prescribed amount. The butterfly valves will always be operated in their wide-open positions or else they will be fully closed; they will not be required to operate at partial openings for regulation purposes.

THE BRAKE

The oil pressure provided by the oil-pump motor for the operation of the butterfly valve is also employed to release the opposing pair of brake disks in the brake cylinder above the rotor. (See Fig. 9.) The outer peripheries of these disks are formed on a taper to engage the tapered friction surfaces in the cover and in the brake cylinder.

Twenty-four powerful coil springs are placed in evenly spaced pockets formed in the faces of the brake disks. These springs force the disks apart and bring the friction surfaces of the disks into engagement with friction surfaces in the cover and brake cylinder, locking the rotor and the butterfly leaf against rotation in either direction. A closely fitted seal ring is inserted into rectangular seal-ring slots in the faces of the brake disks close to their outer diameters, shown in Fig. 9. This seal ring prevents oil under pressure from acting upon the adjacent faces of the brake disks and thus augments the separating force produced by the springs.

The hubs of the upper and lower brake disks are machined to make a close sliding fit in the bores of the cover and brake cylinder, respectively, and each is fitted with three piston rings to prevent the escape of oil. By this construction the oil supplied by the hydraulic-rotor control and entering beneath the lower brake disk will act upon its under face as a piston and lift it away from the friction surface of the brake cylinder, compressing, in so doing, the coil springs between the brake disks. As soon as the lower brake disk has thus been lifted, the oil flows upward around its periphery and thence through ports communicating with the pressure chamber above the upper brake disk, which is then released from its friction seat, and the hydraulic rotor is free to turn.

As soon as the rotor has reached the limit of its travel in either the opening or closing direction, the limit switches shut down the oil-pump motor, the oil pressure in the system drops to zero, the coil springs between the brake disks force the brake disks apart and into engagement with their respective seats, and the rotor and the valve leaf are locked in their new positions.

THE GREASE PUMP

On the right-hand side of the rotor base in Fig. 9 is the end elevation of the automatic, motor-driven, high-pressure, gear-type, grease pump. The pump and motor are also shown in sectional plan *A-A* of Fig. 11. The motor is started simultaneously with the main oil-pump motor. The pump delivers grease under a pressure of 6000 lb per sq in. to the stem bushings during the entire period of rotor movement. This grease pump is provided with a reservoir holding sufficient grease for a number of complete opening and closing cycles.

From Fig. 9 it will be seen that by removing the nuts from the studs through the base flange of the hydraulic rotor, and then uncoupling the two grease pipes running to the stem bushings in the butterfly-valve body, the hydraulic rotor can be lifted off the butterfly valve as a complete unit without further dismantling or draining of the oil within it.

MISCELLANEOUS DETAILS

In designing the hydraulic-rotor mechanism it was essential that it be capable of withstanding an oil pressure of 1800 lb per sq in. when closing under emergency conditions as defined under *B*. As originally designed the stators were made separate from the cylinder in order to facilitate machining and grinding the interior wall surfaces. The stators were to have been bolted to the cylinder and keyed top and bottom to prevent overturning. When this construction was subjected to a mathematical analysis to determine stresses and deformations, many objectionable features were uncovered, with the result that the construction shown in Figs. 9, 10, and 12 was adopted.

Fig. 12 shows the detail construction of the rotor cylinder. It will be noted that 52 dowels, 2 in. in diameter, made of S.A.E. 2340 steel dowels, are provided in the upper flange and as many more in the lower flange. It will also be noted that these dowels are more closely spaced in the radial zones subtended by the stator bases. This was done to take the turning-torque reactions produced by the rotor and to aid in anchoring the stators against the overturning effect, thus relieving the cylinder from distortion which would otherwise be produced by that effect. The castings for these cylinders weigh 22,565 lb each.

When the check sheets for the final designs were completed it was found that all principal stresses resulting from normal operation, as defined under *A* for design purposes, were less than

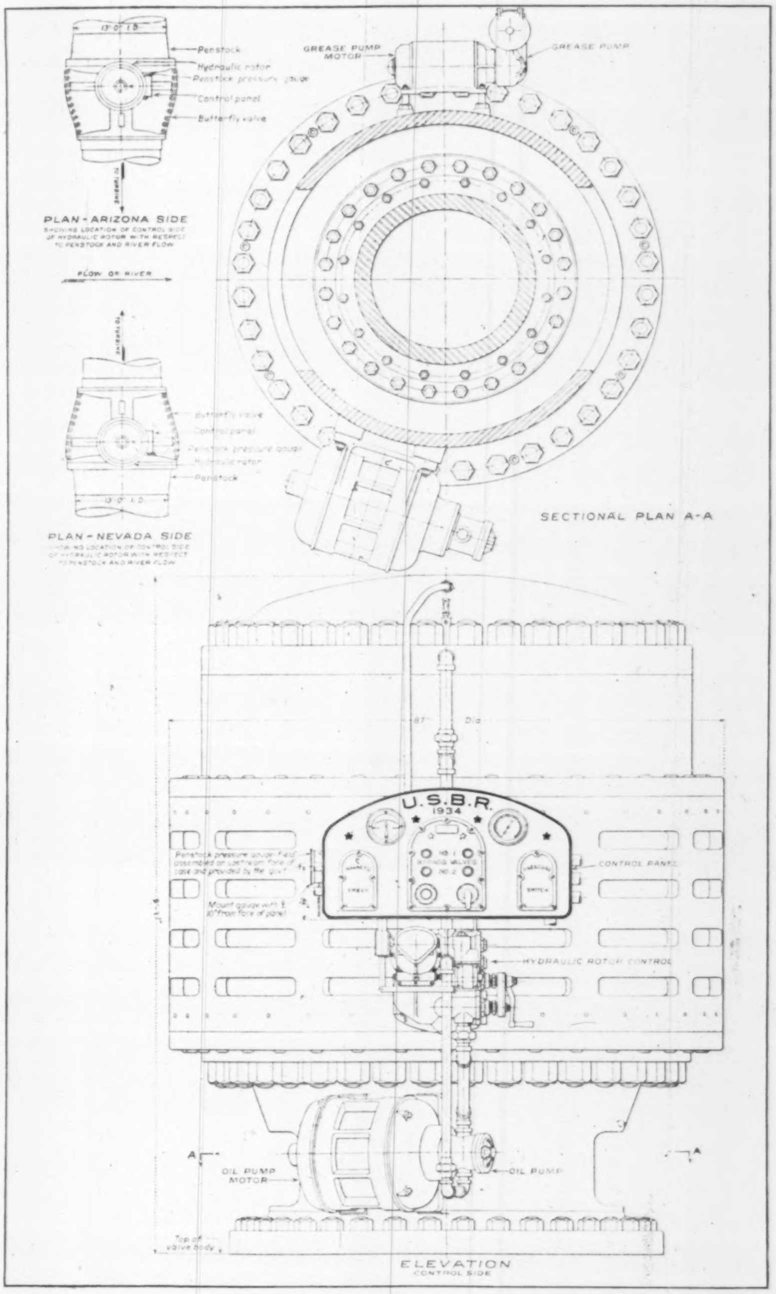

13'-0" I.D.
Penstock
Hydraulic rotor
Penstock pressure gauge
Control panel
Butterfly valve
TO TURBINE
PLAN - ARIZONA SIDE
SHOWING LOCATION OF CONTROL SIDE OF HYDRAULIC ROTOR WITH RESPECT TO PENSTOCK AND RIVER FLOW
FLOW OF RIVER
TO TURBINE
Butterfly valve
Control panel
Penstock pressure gauge
Hydraulic rotor
Penstock
13'-0" I.D.
PLAN - NEVADA SIDE
SHOWING LOCATION OF CONTROL SIDE OF HYDRAULIC ROTOR WITH RESPECT TO PENSTOCK AND RIVER FLOW
GREASE PUMP MOTOR
GREASE PUMP
SECTIONAL PLAN A-A
87" Dia
U.S.B.R.
1934
NO. 1
BY-PASS VALVES
NO. 2
SWITCH
SWITCH
Penstock pressure gauge field assembled on upstream face of case and provided by the gov't
Mount gauge with ℄ 10" from face of panel
CONTROL PANEL
HYDRAULIC ROTOR CONTROL
A
A
OIL PUMP MOTOR
OIL PUMP
Top of valve body
ELEVATION
CONTROL SIDE

FIG. 11 HYDRAULIC ROTOR WITH ITS CONTROL PANEL AND GAGES

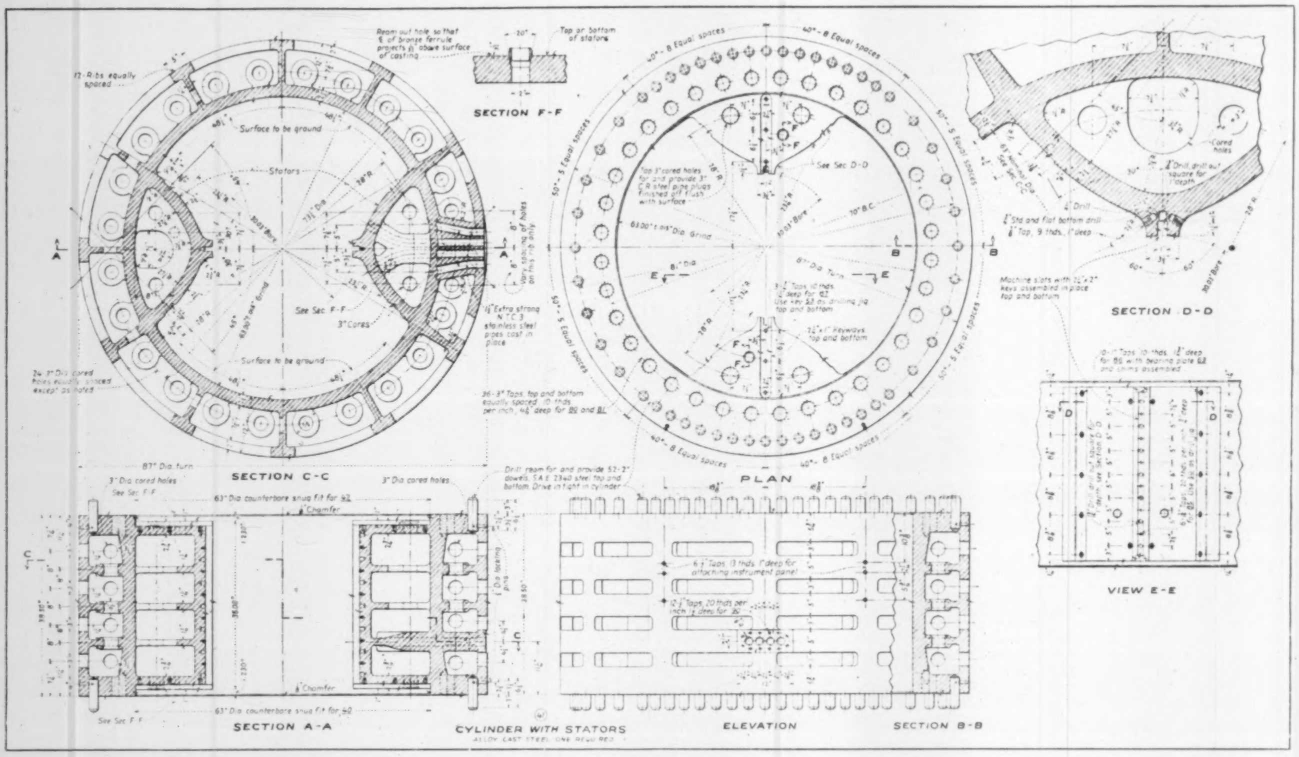


FIG. 12 DETAILS OF THE ROTOR CYLINDER AND ITS STATORS AND THE HEAVY RIBBING AND DOWELING CONSTRUCTION TO HANDLE AN OIL PRESSURE OF 1800 LB PER SQ IN.

one-third of the yield strengths of the materials employed, and for emergency operating conditions as defined under *B* they were well below the value of three-fourths of the yield strengths of the materials used.

The principal dimensions of the hydraulic-rotor mechanisms are 7 ft 3 in. maximum diameter and 9 ft $3^1/_4$ in. overall height. The weight complete of the rotor unit only is 84,000 lb.

The principal dimensions of the butterfly valve proper are 10 ft 6 in. overall length, 15 ft $11^1/_4$ in. greatest breadth, and 19 ft $9^5/_8$ in. overall height. The inlet diameter is 13 ft $9^3/_8$ in. and the outlet is 10 ft 9 in. The weight of the upper half of the valve-body casting is 70,625 lb and of the lower half 75,020 lb, while the leaf weighs 92,875 lb.

The overall height of the butterfly valve and hydraulic-rotor mechanism complete is 28 ft $4^1/_8$ in., while the weight complete is 380,000 lb.

Several features of the butterfly valves described in this paper are believed to be new and novel, and patent applications have been made covering them.

ACKNOWLEDGMENT

The manufacturers of the 55,000-hp and 115,000-hp turbines with which these valves will be used were naturally keenly interested in securing the most favorable hydraulic conditions possible in order that they could successfully meet their guaranteed turbine-performance characteristics. Their representatives and engineers gave much valued counsel and aid in the many problems relating to the shaping of flow contours of the valve bodies and leaves, the bolted flanges joining the upper and lower valve-body castings, and the distribution of stresses from the heavy stem loadings into the valve bodies. The author takes this opportunity to express his thanks on behalf of the Bureau of Reclamation for the cordial cooperation given by W. M. White, manager and chief engineer, and F. Schmidt, engineer of designs, of the Allis Chalmers Co., Milwaukee, Wis.; to O. V. Kruse, sales manager, I. P. Morris Division, Baldwin-Southwark Corp., Philadelphia, Pa.; to D. G. Moorhead, sales manager, and to R. V. Terry, hydraulic engineer, Newport News Shipbuilding and Drydock Co., Newport News, Va.; and to D. J. McCormack, sales manager, S. Morgan Smith Co., York, Pa.; all of whom contributed constructive criticism, friendly advice, and suggestions that were timely.

Government specifications 544 were prepared containing complete detail drawings for the 120-in. valves for use with the two 55,000-hp turbines, and for the 168-in. valves for use with the fifteen 115,000-hp turbines, including drawings for three alternate forms of construction, namely (1) with leaf, body, and cylinder of cast steel, (2) with leaf, body, and cylinder of welded plate steel, and (3) with leaf and body of composite alloy cast steel and welded plate-steel and cylinder of welded plate steel.

The low bid for the alloy cast-steel leaf, body, and cylinder was considerably less than the low bids for either of the other two alternate designs and award was made for four 168-in. and one 120-in. valves, complete with all accessories, to the Hardie-Tynes Mfg. Co., of Birmingham, Ala. Fabrication is now well advanced.

Larger butterfly valves than those described herein have been built and are in service, notably the 27-ft valves at Conowingo, which, however are under quite low head, namely 90 ft, with a maximum static load of 3,220,000 lb, while the 14-ft Boulder valves, when subjected to maximum working pressure, will have a static water load of 6,500,000 lb. It is believed that there are no valves as large as these being fabricated for Boulder dam which are subjected to as heavy a working pressure or as severe emergency operating conditions as are these valves.

Part III of this paper, which will describe the needle valves and the Paradox emergency gates, will appear in a later issue.

THE STRANDGREN CYCLOGIRO

Description of an Unusual Type of Aircraft in Which Rotating Wings Provide Both Lift and Propulsion

By C. B. STRANDGREN[1]

ALTHOUGH AVIATION is still young, some aeronautical authorities already believe exclusively in the airplane, which, according to their opinion, is based on the soundest principle and has the greatest possibilities of future development. Aside from the harm likely to be done to aeronautical progress by this opinion, the belief itself is not easy to justify.

By reason of its very nature, the airplane will never be able to travel below a certain minimum speed—the stalling velocity—nor will it be able to land and take off in vertical flight. Because of this minimum landing and starting speed, aviation fields must be made sufficiently large and smooth to permit safe operation, and pylons, buildings, chimneys, and electrical transmission lines in their vicinity present constant hazards, particularly in bad weather.

It is obvious that a new type of aircraft, with the same efficiency as the airplane and with the possibility of reducing its speed at will down to zero and of landing and taking-off vertically or at any desired angle, would represent an important development in aviation. The principle of such an aircraft exists and has been practically tested to demonstrate that it is both aerodynamically and mechanically satisfactory.

The author has studied and worked on the principle since 1924, but because of his limited financial means, material progress has been slow. Notwithstanding this handicap, the invention is now ready to enter the commercial stage of its development.

The Strandgren cyclogiro requires a shorter and a lighter fuselage and smaller undercarriage than a conventional airplane. The novel feature of its design lies in the elimination of the conventional wings and their replacement by a system of rotating blades, which provides not only the entire lift but also the entire propulsion.

The rotating wing, or wheel, not unlike a paddle wheel, turns about a central transverse shaft passing above or through the fuselage. This wheel consists of a number of blades distributed symmetrically around the central shaft. Each blade is mounted on anti-friction bearings located at two or more places along the span, so that it can swing about the axis passing through these bearings. Two links connect each blade with an oscillation control as shown in Fig. 1. The wheels may be located on each side of the fuselage, which contains the motor as shown in Fig. 2, and are driven by means of a power transmission with a free-wheeling clutch. The torque reaction of the engine drive is balanced by the weight of the fuselage with its motor, tanks, and load on the wheel shaft.

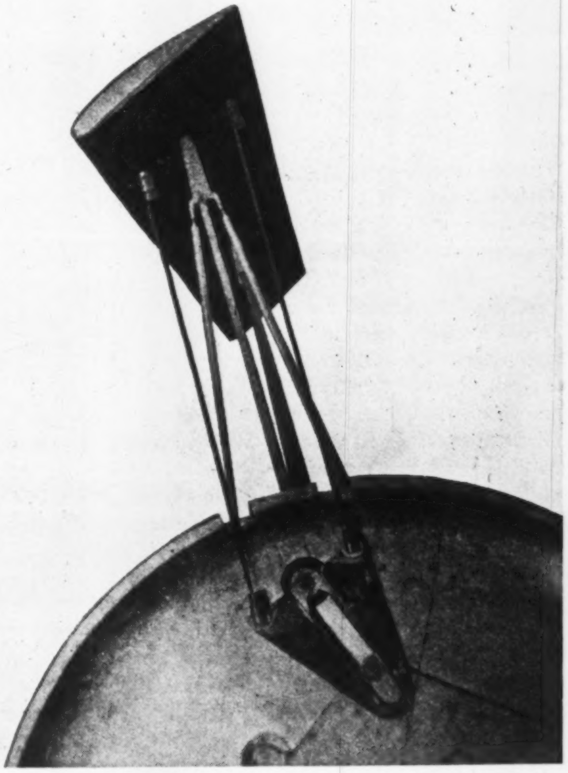


FIG. 1 LINKAGE BETWEEN BLADE AND OSCILLATION CONTROL

KINETIC AND AERODYNAMIC PRINCIPLES OF THE CYCLOGIRO

If the cyclogiro is moved along a straight path, that is, if the central shaft has a motion of translation relative to the air stream, the path followed by any one of the rotating blades is a combination of translation and rotation about the shaft, which makes it a cycloid whose rolling circle has a diameter depending on the speed of rotation of the wheel and the velocity of translation of the shaft relative to the air stream. Denoting the speeds of translation and rotation and their component by V, U, and W, respectively, three different cases are possible (see Fig. 5), according as U is greater than, equal to, or less than V. With the tangents to these paths the blades must form suitable and varying angles in order to produce most economically the forces whose vertical and horizontal components make up, respectively, the lift and the pull of the wheel (Fig. 3).

The instantaneous axis of rotation is situated at a distance V/ω from the axis of the wheel. This distance is by definition the radius r of the rolling circle. Every point in the plane of the wheel except its center is tracing a cycloid in the fixed plane. The normals to these cycloids, at the moment considered, pass through the instantaneous axis. The distance from this axis to a blade is the instantaneous-radius vector ρ, and the instantaneous velocity of the blade is $\omega\rho = W$.

If the chords of the blades coincide with the tangents to their paths, the normals to the blades coincide with the radii vectors. (See Fig. 4.) If the wheel advances through a solid medium, the angle of incidence must be zero so that each blade may glide along its path; this implies that the normals to the blades coincide with the instantaneous radii vectors and consequently all pass through a point coinciding with I. Fig. 4 shows a wheel with four blades whose normals pass through the point I. If this wheel were rotated with the angular velocity ω it would necessarily advance along the directrix D with the velocity $V = \omega r$, and each blade would follow indicated cycloids. It appears that the velocity V and direction of displacement D depend solely on the position of I relative to the center of the wheel and on its angular velocity.

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[1] Neuilly-sur-Seine, France.

The example of the wheel working in a solid medium can be replaced by that of a weightless aircraft in an ordinary fluid. In both of these cases the angles of incidence of the blades with their paths are zero and the normals to the blades are to be controlled in such a way that they all pass through a single point I which determines the instantaneous axis.

In the case of an ordinary fluid, assume that the wheel in Fig. 4 is traveling along D with a speed whose horizontal and vertical components are V and v, respectively, with the converging point I always remaining in the same position. Assume also that as the wheel turns, the fluid is yielding vertically with a speed equal to v. In this case the direction of flight will be horizontal along the new directrix D' with the velocity V'. Under these conditions the instantaneous axis will no longer pass through the convergence point I of the normals but will pass through I' at the distance r' from the center. The blades now form angles of incidence with their paths determined by the "yielding velocity," that is, by the position of the instantaneous axis I' relative to the convergence point I of the normals, and their new paths are determined by the rolling of the circle of radius r' on the directix D'. The effect of the horizontal yield or slip of the fluid is of minor importance and slightly reduces r' and the horizontal speed.

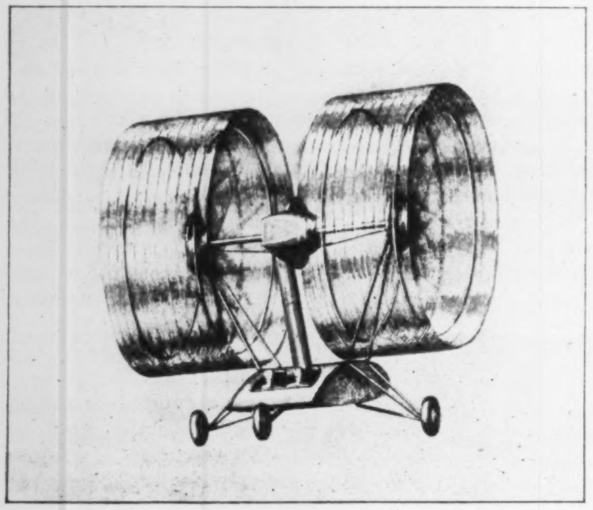

FIG. 2 FIRST STRANDGREN CYCLOGIRO WITH A WHEEL ON EACH SIDE OF THE FUSELAGE

In aerodynamics the vertical and horizontal yieldings of the fluid are called "downwash" and "slipstream," respectively, and their values depend on the circulation around the airfoils and on the free vortices in the fluid. The velocity of the downwash and the slipstream is termed the "induced" velocity, and its value in different parts of the fluid can be calculated when the circulation and its rate of variation are known. In the case of a single airfoil with constant circulation this calculation is easy, but in the case of a cyclogiro with several blades and variable circulation it is very complicated.

Now a very important question may be asked: What should be the value of this induced velocity in different parts of the fluid in order that the drag may be a minimum or the efficiency a maximum? This problem was solved long ago by Munk and Betz, and the solution is that in the wake far behind a wing, propeller, or the like, the induced velocity should have the same (constant) value along the span, or, in other words, the fluid must be moved as though it were a solid.

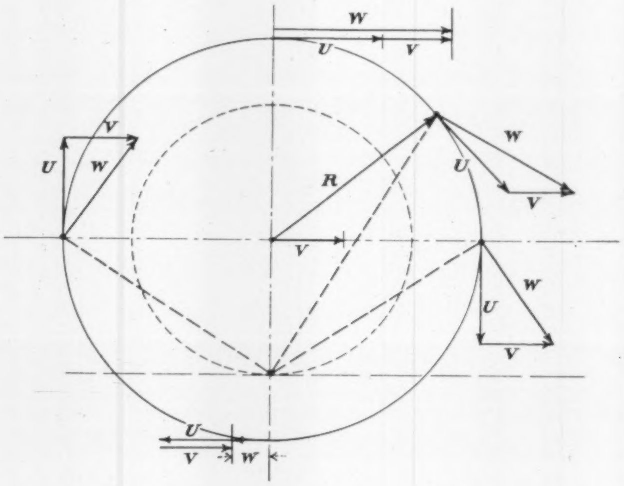

FIG. 3 DIAGRAM SHOWING SPEED COMPONENTS

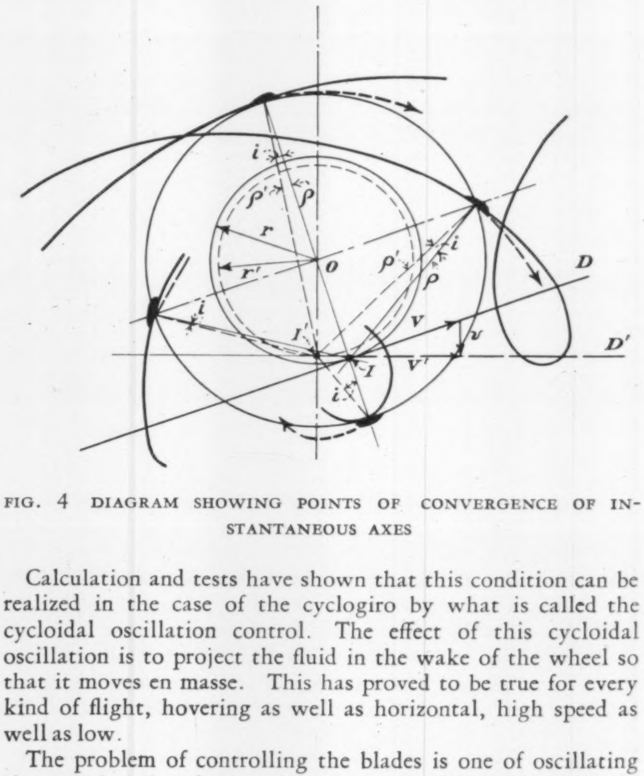

FIG. 4 DIAGRAM SHOWING POINTS OF CONVERGENCE OF INSTANTANEOUS AXES

Calculation and tests have shown that this condition can be realized in the case of the cyclogiro by what is called the cycloidal oscillation control. The effect of this cycloidal oscillation is to project the fluid in the wake of the wheel so that it moves en masse. This has proved to be true for every kind of flight, hovering as well as horizontal, high speed as well as low.

The problem of controlling the blades is one of oscillating them so that they form such angles with their paths that for a given thrust or air force the power is a minimum. The number of variables depends on the state of development of the problem and the character of the mechanism. In the case of the cyclogiro the minimum is two parameters—the direction and the speed of flight.

Three different forms of cycloid must next be examined. In the first case (see Fig. 5), when V, the speed of translation, is less than U, the speed of rotation, the form is that of a curtate cycloid and the convergence point of the normals to the blades lies within the circle described by the axes of the blades. Relative to the aircraft the blades are oscillating only, whereas in the other cases, as will be explained, they are rotating about their supports. The main factors in this case are: (1) The blade is easily supported along its span by roller bearings as it does not rotate; and (2) the traveling speed V can be reduced to zero whatever the value of U may be. On account of this feature, this type of cyclogiro is the only one with a traveling speed between zero and a certain maximum, slightly less than U.

In the second case when $V = U$ (Fig. 5), a cycloid proper results and the convergence point of the normals to the blades lies on the circle described by the axes of the blades which now coincides with the rolling circle. It is easily seen that the blades are rotating relative to the wheel with an angular velocity $\omega/2$ and in a reverse direction. The result is an aircraft that, although it is a cyclogiro, has exactly the same characteristics of stalling speed as an ordinary airplane. The construction is complicated because the blades revolve. On account of the speed of revolution $\omega/2$, the positions of the blades change every other revolution, the leading edge becoming a trailing edge and vice versa, so that their cross-section must be entirely symmetrical about the main axis, and this results in a defective blade profile.

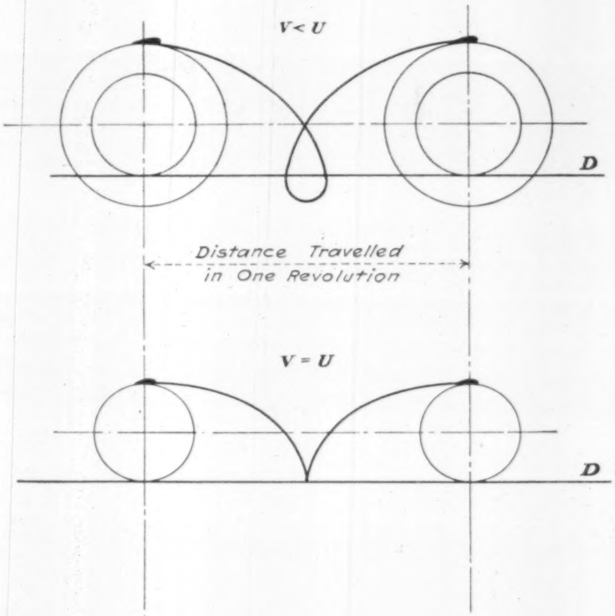

FIG. 5 DIAGRAM SHOWING RELATION BETWEEN TRANSLATION AND ROTATION OF A CYCLOID

Lastly, a word concerning the third case. When V is greater than U (Fig. 5), the paths of the wings are prolate cycloids and the instantaneous axis lies outside of the wheel. The blades are revolving relative to the wheel and oscillating relative to the aircraft, which leads to a construction similar to that of the second case. The traveling speed can be varied within a great range of values but all the speeds are greater than U. An aircraft provided with a wheel of this type would have a take-off speed comparatively higher than that of an airplane.

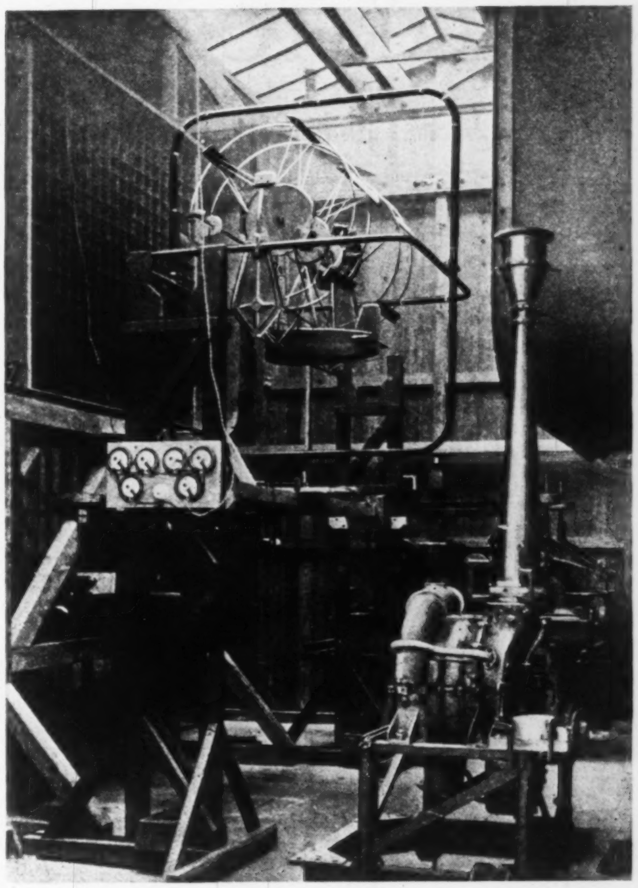

FIG. 6 LABORATORY INSTALLATION FOR TESTING STABILITY OF THE WHEEL MODEL

Only the second of the last two cases, to the author's knowledge, has been tested and this sometimes with the intention of creating an aircraft having characteristics that can only be found with a cyclogiro of the first type. The second case is the least interesting, because of the very limited range of available speed and the incorrect blade profile. The first and third cases make use of customary blade profiles and both have large ranges of speed.

Returning to the wheel, which is the subject of this paper, it has been shown that, by changing the location of the convergence point I of the normals, it is possible to fly the cyclogiro in any direction and at different speeds. If I is placed on the horizontal line through the center of the wheel on its right, purely vertical flight or hovering results. Although this aircraft can fly in every direction at a great range of speed it has, naturally, a most economical specific speed for each direction.

The value of this speed in each particular case is determined by certain variables, such as size of the wheel and the blades, the total weight, and the speed of rotation U. Its speed in hori-

(Continued on page 567)

A GEAR-TOOTH *Designing* MACHINE

A Quick, Accurate Method of Drawing Generated Tooth Shapes

By HERBERT S. INGHAM[1]

SINCE THE development of our present generating methods of gear manufacture, it is no longer necessary to draw the shapes of gear teeth. When gear teeth were cast on a wheel, their shapes had to be accurately drawn for the patternmaker; and when gears were cut by the form-planing process, the templet had to be accurately laid out. Now, however, with modern gears manufactured by the generating processes, the necessity for this sort of drafting has been eliminated. A gear which is to be generated is completely determined by the tool and by the settings of the gear-generating machine. The first reaction of the engineer to this change has been to consider his problem simplified. This simplification exists, however, only if we are to restrict ourselves to the same limited number of tooth forms as were used in the templet days.

The advent of the generating method of gear cutting has given us many advantages, one that is most outstanding and least used being the flexibility of design that is permitted. The importance of designing gear teeth to meet the conditions of their particular application is demonstrated by the extensive use today of specially designed teeth for the more severe applications. The value of designing gears for their job, as other machine parts are designed, is so well established that it needs no long discussion here; and the ease with which specially designed gears may be produced without additional time or cost of production brings up the question as to why this practise is not more universally applied. It may be that too many engineers are willing to accept the simplification in layout work brought about by the generating method without accepting obvious advantages that should come with it. One has only to look at representative industrial gears and note the many undercut teeth there are to realize that we are slow to make the advantages of modern tooth design universal.

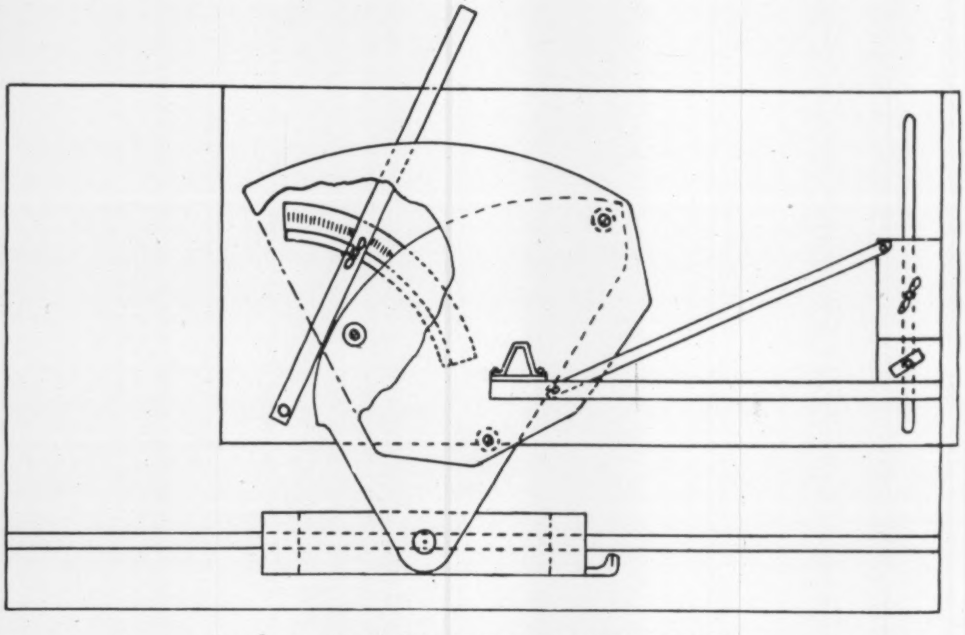

FIG. 1 ELEMENTS OF GEAR-TOOTH DESIGNING MACHINE

THE ELEMENTS OF THE DRAWING MACHINE

The generated-gear-tooth drawing and designing machine is described here with the hope that it may prove of some value in simplifying tooth design, and also in the hope that in some cases it may prove conducive to a more complete design than would otherwise be attained. No machine can be a substitute for thought, but a calculating machine can save time and energy for thought. This gear-designing machine is largely a calculating machine. It will not obtain any results that could not be calculated. It does, however, produce results that would not be calculated; and that is the key to its purpose. It offers the designing engineer a quick and accurate method of drawing any generated tooth shape. The drawings not only furnish dimensions of the teeth at any and all points but also show such things as undercut, its extent, and by what adjustment it may be eliminated. They show the place at which adjusted addendum teeth become too narrow or pointed, and, most important of all, they furnish the designer with concrete examples upon which to base his judgment in regard to the proportional shapes of two mating teeth, whether they be of the conventional unequal-addendum design or of a special pressure-angle design.

The machine is very simple in its construction and operation. It draws a gear-tooth form by generating it on paper in a manner similar to that in which a gear tooth is generated in a hobbing machine. A drawing board, upon which the paper is fastened, is caused to move with a combined rotary and translatory motion beneath a marker which is shaped to represent a tooth space from the basic rack of the tooth system being used. The motion between the paper and the marker represents the motion which takes place between the gear blank and the hob. Thus it is possible to transfer to a desk the essentials of the gear hobber and "cut" as many trial gear-tooth shapes as desired in a very short time. To facilitate its use with a hobbing machine, the designing machine has the same adjustments and works

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[1] Gear Research Engineer, Sterling Electric Motors, Inc., Los Angeles, Calif. Jun. A.S.M.E.

Presented at a meeting, Los Angeles, Calif., Feb. 13, 1934, of the Los Angeles Section of The American Society of Mechanical Engineers.

from the same "known" quantities as the hobber. The marker, which represents a tooth space from the basic rack (or hob), is made considerably larger than the actual pitches used. This produces drawings of an enlarged scale, the actual scale of the drawing depending upon the pitch of the gear being represented. For instance, if the marker is of a $1^{1}/_{2}$-diametral-pitch size, it will draw teeth of a 15-pitch gear with 10 to 1 enlarged scale and teeth of a 12-pitch gear with an enlarged scale of 8 to 1. Thus, as the proportions of the basic rack of a tooth system are always the same for that system, only one marker is needed for the designing machine to draw all tooth numbers and pitches within the limits of its operation.

The gear-tooth designing machine is shown in its most elementary form in Fig. 1.

To produce the desired relative motion for drawing the tooth forms, the drawing board must both slide and rotate beneath the marker. This motion can best be understood by imagining a rack to be fastened lengthwise across the machine so that the marker will form one of its tooth spaces. If the gear which is to have its tooth form represented by the marker were placed on the movable drawing board with its center at the sliding pivot, then it would mesh with and roll along the imaginary rack and produce the desired motion between the drawing board and the marker. As tooth forms of gears having different numbers of teeth are to be drawn, this motion must be adjustable so as to give varying amounts of rotation of the drawing for a given amount of sliding of the pivot. For instance, when drawing a tooth of a 10-tooth pinion, the drawing board must rotate one-tenth of a revolution or 36 deg while the center of the gear at the pivot moves longitudinally a distance equal to one circular pitch of the marker. When drawing a 36-tooth gear the drawing board would rotate only 10 deg while moving the same circular-pitch distance longitudinally. The gear-tooth designing machine attains this motion by the use of an involute cam which makes contact with a straight edge that is adjustable with respect to its angular position. The angular adjustments of this straight edge provide an infinite number of ratios between the rotary and translatory motion. It might be well to mention here that there is no direct connection between the fact that the cam is an involute and that teeth drawn are usually involutes. The cam and straight edge produce the required motion only. Any tooth form can be drawn by using a marker that represents the desired basic rack. The cycloidal tooth would be drawn by using a cycloidal rack-form marker. Splines of a spline shaft may be drawn by using a marker shaped like a tooth space from the spline hob.

The physical dimensions of the machine determine the largest diameter gear that can be represented. For this reason a smaller marker is used for the larger tooth number. With a drawing board of about 13 in. radius, the largest $1^{1}/_{2}$-pitch gear that could be drawn would have 36 teeth. A $4^{1}/_{2}$-pitch marker, exchanged for the original marker, would draw tooth numbers from 36 to 108 with a scale equal to one-third of the original scale. The angular settings of the straight edge would start back at zero and progress over the same range as before. Thus it is possible for a small compact machine to draw teeth covering as wide a range as necessary while the more critical shapes of the smaller pinion teeth are represented to the larger scale.

The use of the gear-tooth designing machine is facilitated by charts giving the pitch diameter and outside diameter for gears of the marker's pitch. The block slider which locates the position of the marker is provided with a screw movement and vernier scale (not shown in the illustration) for more convenient and accurate operation. In causing the marker to draw on the paper, it has been found very satisfactory to use so-called "metallic" paper such as used for engine indicator-card diagrams. The marker is of brass and therefore draws on the paper directly wherever it rubs.

DRAWING A TOOTH FOR A PINION

Suppose we are to draw a tooth for a 16-tooth spur pinion of 12 diametral pitch, $14^{1}/_{2}$-deg involute. The larger (1.5 diametral pitch) marker is used, the resulting drawing being enlarged eight times. The straight edge is set to the "16-tooth" mark and metallic paper fastened to the drawing board. A line representing the outside diameter of the pinion blank is then drawn on the paper with compasses, the point of the compass being placed in a punch mark provided in the center of the sliding pivot. The cam slider is "centralized" (by the scratch lines) and the marker brought to just touch with the outside diameter of the pinion. The vernier is set at zero and the marker moved into the "blank" a distance equal to the total depth of the cut. One movement of the slider along the track now generates the desired tooth. This tooth is shown in Fig. 2. The whole process is exactly analogous to that of gear hobbing. Nothing has to be known about a gear to draw it that does not have to be known to hob it. Even the pitch diameter need not be calculated. It follows that information obtained by the machine in design work applies directly to the cutting of the improved gear. This point is emphasized because time is an important item, and, if any additional information had to be obtained for the drawing machine, part of its purpose would be defeated.

DESIGNING HELICAL GEARS

As helical gears are becoming popular because of their greater smoothness and quietness of operation, no discussion of tooth-form design is complete that does not consider helical teeth. In cutting a helical gear, the hob is generating the teeth at an angle to the cylindrical blank of the gear. If an imaginary section were cut through the gear and hob perpendicular to the helical teeth, this section would show the hob teeth as the basic rack, and it would show the gear blank as an ellipse. This ellipse would be the result of the intersection of a plane with a cylinder, the plane being at an angle equal to the gear helix angle from a normal section of the cylinder. The hob will be generating the teeth on the minor axis of this hypothetical ellipse. Now the radius of curvature of the ellipse at the minor axis will be equal to the radius of the cylinder divided by the cube of the cosine of the helix angle. As the pitch-circle radius is considered it will be seen that the generating action for a helical gear will be similar to that of a spur gear with a larger pitch diameter. This results in the formula that for helical gears the "equivalent" number of teeth equals the actual number of teeth divided by the cube of the cosine of the helix angle. This formula has to do with the shape of the teeth and is used for selecting milling cutters for cutting helical gears by that method.

On the gear-designing machine normal sections of helical gear teeth are drawn just the same as spur-gear teeth by using their "equivalent" number. A 16-tooth 23-deg helical gear would be drawn the same as a 20.51-tooth spur gear. Because of the fractional number of teeth, it is apparent that some device such as the cam and adjustable straight edge is necessary to obtain all ratios of movement. By calibrating the straight-edge protractor directly in real tooth numbers for popular helix angles, and by having charts giving the outside diameters directly from the real tooth numbers for these "equivalent" gears, helical teeth are just as easily drawn as spur-gear teeth. Fig. 3 shows a standard 16-tooth $14^{1}/_{2}$-deg pressure angle 23-deg helical pinion tooth for comparison with the spur tooth in Fig. 2.

The undercut produced in these pinions is at once apparent. The helical pinion shows a better condition than the spur pinion but neither is good design practise. Besides the weakening effect on the tooth strength, the undercut has reduced the active involute profile by about one-third. Fig. 4 shows an adjusted tooth for a similar 16-tooth spur pinion cut with the same hob set at the same depth from the outside of the blank.

The only difference is in the diameter of the blank. The radius has been increased by 56 per cent of the addendum height. This pinion may or may not run with a $14^{1}/_{2}$-deg pressure angle on standard center distances, according to the design of the gear with which it meshes. The commonly used "unequal addendum" method would be to reduce the gear-blank radius by the same amount as the pinion was increased. The gears would then run on the same center distances as standard gears. This is a very satisfactory way to make the adjustment where the number of teeth in the mating gear is relatively large. If the gear has too few teeth, however, the reduction of its size will cause it to be undercut, producing the disadvantage that we are trying to eliminate in the pinion. Wherever such an adjustment is to be made the gear should be investigated from this standpoint.

SPECIAL TOOTH FORMS

The old principle that standard center distances or "interchangeability" require the use of standard tooth forms is certainly obsolete. Gears with specially designed teeth cannot only be made to run on standard center distances but on any center distances whatever. The tooth number may be changed from standard either as one means of fitting special gears to existing centers; or as a method for changing the transmission ratio to some value not otherwise attainable. Helix angles may be changed or spur gears replaced with helical ones; pitches may be changed on existing machines or a machine may be built to utilize a number of different pitches for different ratios and load requirements. This real freedom of design makes the standard tooth "interchangeability" seem restricted indeed.

To cite a practical example of the use of such design freedom, an actual case will be considered. Recently it was found necessary to obtain an increase ratio to give 3660 rpm at the high-speed shaft of a 20-hp speed increaser when the motor ran at 1480 rpm. This required a gear ratio of 1 to 2.47. The gear case to be used was designed for a maximum of 30 hp. The regular gears for this case were 8-pitch 23-deg helical gears and were used in combinations with the total number of teeth in the pinion and gear equal to 54. The center distance was such as to allow for an adjustment of the tooth shape. Now the desired ratio could not be obtained with any combination of 54 total teeth, but it could be obtained with a 66-tooth combination by making the pinion with 19 teeth and the gear with 47. As only 20 hp was required, it was possible to reduce the size of the teeth to 10 pitch and still have more than ample strength. By arbitrarily changing the helix angle to 27 deg, a "standard" center distance for the new 10-pitch 66-tooth set was obtained that was slightly smaller than the actual centers of the gear case. The 19-tooth pinion was then cut oversize so as to eliminate undercut and so as to make the actual center distances of this new set just fit the existing centers of the gear case. Because of the purposely increased center distance of this combination the gears operated at a pressure angle of 16 deg 24 min, although they were cut with a standard $14^{1}/_{2}$-deg hob. There was nothing "makeshift" about the final design. The strength was adequate and the changes in pitch and helix angle were in the direction of increasing the tooth overlap. By this means it was not only possible but very practical to obtain the desired speed of 3660 rpm while with the original gear combination the nearest attainable speed would have been 145 rpm too slow.

These are not new ideas. The attempt is only to emphasize the old argument that better advantages should be taken of the generating principle of gear manufacture. The design problem cited is only one special case illustrating only one particular advantage. It does illustrate, however, what amounts to a real necessity for special design in practical work. Specially designed teeth do not cost more to produce as standard hobs are used. The objections must be that they are more trouble to design.

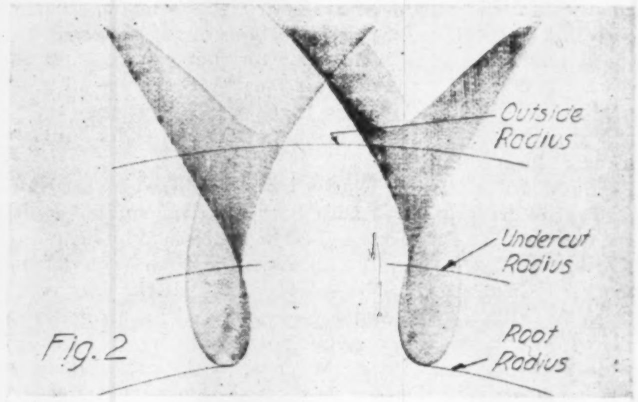

Fig. 2

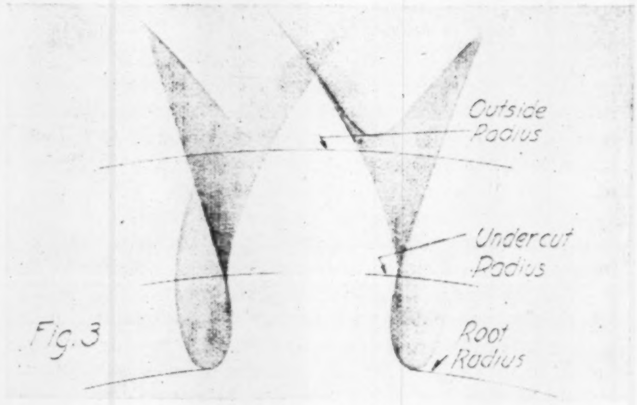

Fig. 3

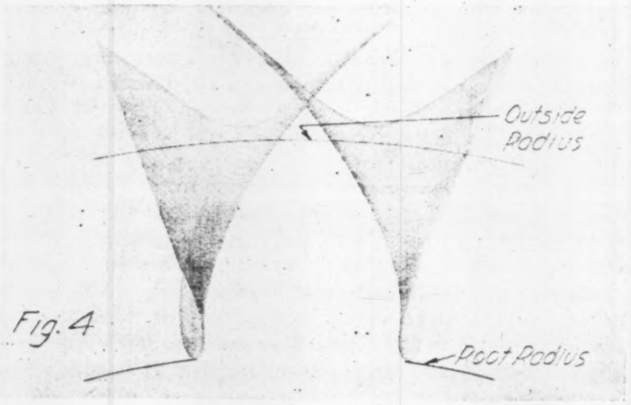

Fig. 4

UNEQUAL-ADDENDUM TEETH

One encouraging step in the right direction is the popular use of a "rule of thumb" for making unequal-addendum teeth. "Increase the pinion and decrease the gear-blank radius by an amount equal to one-third and the standard addendum height for all cases where the pinion has less than 32 teeth." This rule, like most rules of thumb, fits an average case very well, but certainly should not be taken literally to mean all cases. Fig. 5 shows a standard 13-tooth spur pinion. It is obvious that some adjustment is necessary. Suppose we consider this rule-of-thumb method. Fig. 6 is the 13-tooth pinion adjusted in this way with an increase in blank radius of one-third of the addendum. The undercut has not been eliminated. Fig. 7 is a 24-tooth gear, with its addendum reduced one-third, which might have been intended to run with the 13-tooth pinion. They would not run together with continuous action. This gear would have been undercut if it were cut as a standard tooth, and the extra depth of cut required by the conventional unequal-addendum correction has completely spoiled it. Of course, this is an exaggerated case, but it demonstrates the sort of trouble that may result from blind use of a rule. Figs. 8 and 9 show a 13-tooth pinion and a 24-tooth gear, respectively, that can be cut with the same hob and that will mesh with continuous action. The very slight cutting seen at the bottom of these teeth does not affect any of the otherwise active profile of the teeth, being below the last point of contact.

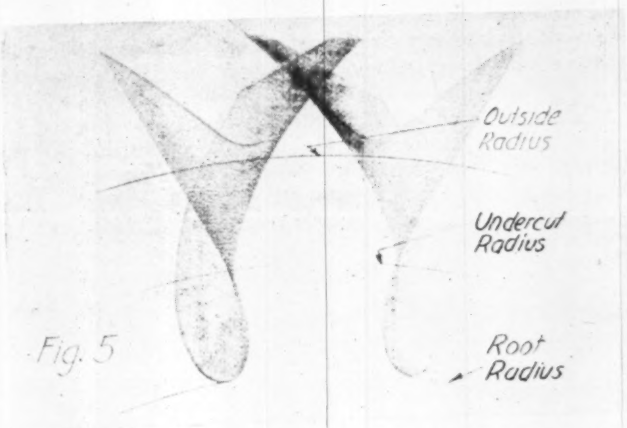
Outside Radius
Undercut Radius
Root Radius

Fig. 5

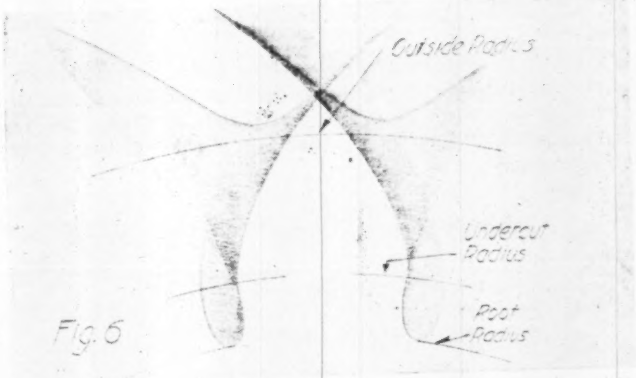
Outside Radius
Undercut Radius
Root Radius

Fig. 6

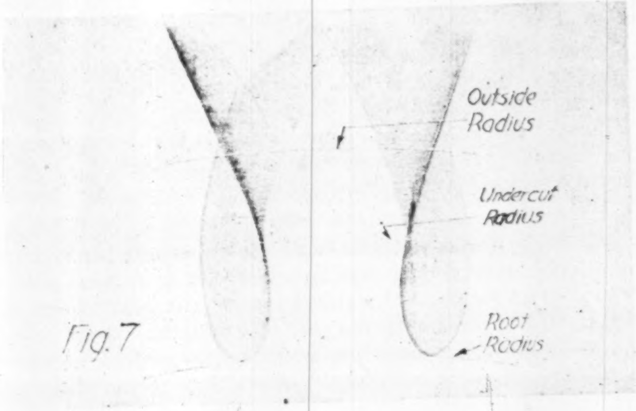
Outside Radius
Undercut Radius
Root Radius

Fig. 7

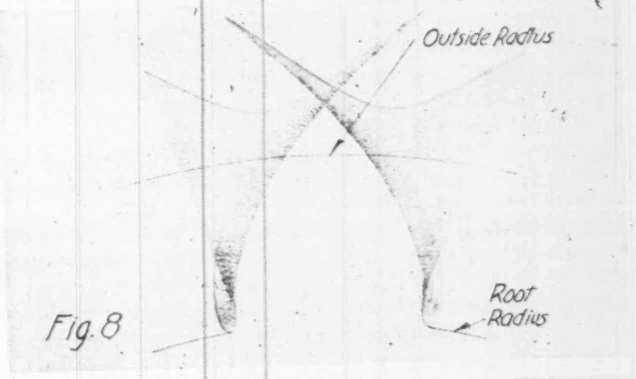
Outside Radius
Root Radius

Fig. 8

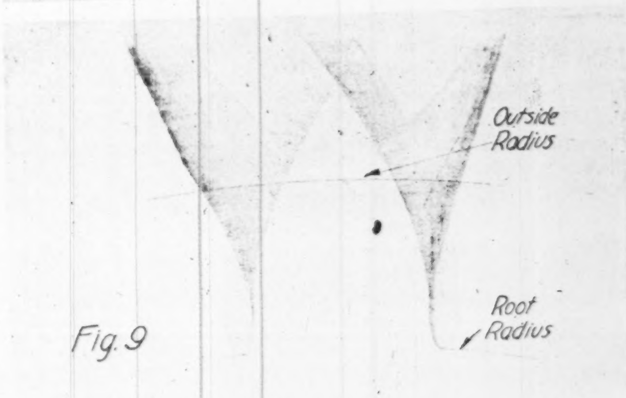
Outside Radius
Root Radius

Fig. 9

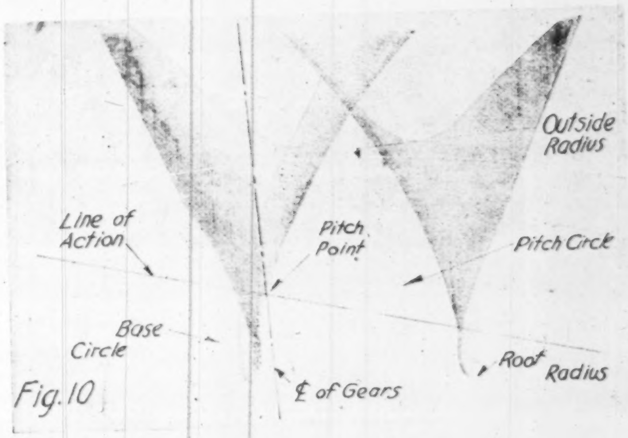
Outside Radius
Line of Action
Pitch Point
Pitch Circle
Base Circle
Root Radius
℄ of Gears

Fig. 10

The tooth shown in Fig. 10 was specially designed for an 18-tooth 23-deg helical pinion of a 10,000 rpm speed-increaser unit. The pressure angle at which it operates is 16 deg 16 min, although it was cut with a standard $14^1/_2$-deg hob. As the original gear is of 16 pitch, this drawing shows the tooth enlarged $10^2/_3$ times. The centerline of the gears, the line of action, and the pitch circle have been drawn in in addition to the outside, base, and root circles, so as to show how it meshes with its mating gear. The last active point on the involute profile is somewhat above the base circle, so that the very critical curve next to the base circle is not used. Slightly more (about threefifths) of the action takes place on the approach side of the pitch point, or on the profile above the pitch circle of the tooth shown, because in this case the pinion is the driven member. Some authorities consider the wear to be greater on the approach than on the recess side; however, in this particular design the gears were overdesigned from the standpoint of strength and wear and the greatest emphasis was put on quietness. For this reason the gears were designed to mesh with nearly two teeth in contact through any section normal to the teeth, besides, of course, the overlap due to the helix. This produces a quiet, smooth-running gear train, as the effects of tooth deflections and small cutting errors are largely canceled out.

In advanced design work of this type the proposed gears would no doubt be layed out diagrammatically as usual, and the important quantities, including blank and cutting dimensions, specific sliding, etc., calculated. The gear-designing machine, however, offers a quick trial-and-error method of preliminary study of a problem; a quick, accurate, and positive check of calculated work; and, what is most important of all, it gives the designer a clearer conception of his work to help in the exercise of his judgment.

DERIVATION OF VARIABLE CAM MOTION

The action of the cam in the designing machine is that of a constant-rise cam with the adjustable feature of the straight edge added to provide variations in the ratio. In Fig. 11 the involute curve representing the active edge of the cam is shown in two positions in contact with the straight line *XY*, which represents the adjustable straight edge. The base circle of the involute is shown in the two positions corresponding to the positions of the involute. On the designing machine the center of this base circle is at the pivot of the cam on the slider and therefore moves in a straight line, shown as *XZ* in Fig. 11. Let *A* be the initial position of the pivot and *A'* be its final position after the slider has moved a distance *d* along the tracks. The slider is set at an angle *x*, with a line normal to the tracks. The generating line *BC* is normal to the curve at *B* and is therefore normal to *XY* when *B* is the point of contact. *B'C'* is normal to *XY* for the final position when *B'* is the point of contact. *B'C'* is then parallel to *BC* and the radius vectors *AC* and *A'C'* are parallel to each other and to *XY* as they are perpendicular to the generating lines. Angle *A'AE* is therefore equal to *x* where *AE* is normal to *A'C'*. By representing the vectorial angles for the two positions of the involute by ϕ and ϕ' we have a measure of the angular positions of the involute; the two reference lines *AC* and *A'C'* being parallel. The angle through which the cam rotates when its pivot moves a distance *d* is $\Delta\phi$ or $\phi - \phi'$. By the definition of the involute, ϕ equals *BC* divided by the radius of the base circle *r;* for *BC* equals the length of the arc subtended by ϕ.

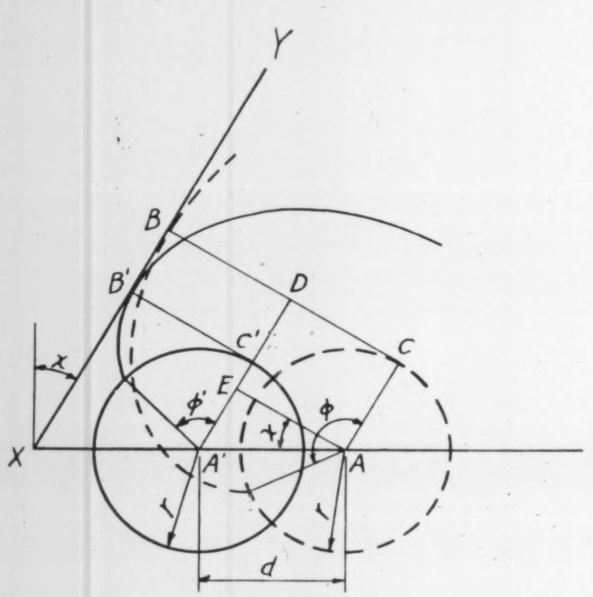

FIG. 11

Hence

$$\phi = BC/r \qquad \phi' = B'C'/r$$

$$\Delta\phi = \phi - \phi' = \frac{BC - B'C'}{r} = \frac{DC}{r} = \frac{EA}{r} = \frac{A'A\cos x}{r} = \frac{d\cos x}{r}$$

Therefore

$$\frac{\Delta\phi}{d} = \frac{1}{r}\cos x$$

As *r*, which is the radius of the base circle from which the active involute edge of the cam was generated, remains constant, the ratio between the rotary and translatory motions of the cam varies as the cosine of the angle at which the straight edge is set.

The cam of the gear-tooth designing machine is determined by the largest marker which is to be used and by the smallest pinion to be drawn. For instance if 12 is to be the limiting number of teeth for a spur pinion and if the marker is to be $1^1/_2$ diametral pitch, then

$$\Delta\phi = {}^1/_{12}\text{ circle} = \frac{\pi}{6}\text{ radians}$$

$$d = \text{one circular pitch} = \frac{\pi}{1.5}\text{ in.}$$

$$\cos x = 1\ (x = 0 \text{ for limiting case})$$

$$r = \frac{d}{\Delta\phi}\cos x = \frac{\pi \times 6 \times 1}{1.5 \times \pi} = 4\text{ in.}$$

The cam would be constructed with a base circle of 4-in. radius. Larger numbers of teeth could then be represented by properly adjusting the straight edge. For convenience, the protractor is calibrated directly in tooth numbers so that in the operation of the machine it is only necessary to set the straight edge to the marks representing the number of teeth in the gear to be drawn.

We might just as well command the sun to stand still as to say that science should take a holiday. Science has turned scarcity into plenty. Merely because it has served us well is no reason why we should charge science with the responsibility for our failure to apportion production to need and to distribute the fruits of plenty equitably. That failure we must charge squarely to organized society and to government. We need economic machinery corresponding to our scientific machinery in precision, in power, and in delicacy of adjustment. Science has done the first job, and done it magnificently. It has shown us how to produce. Now it must show us how to distribute what we produce. It must go forward and not back. To production science we must add economic science, without for a moment ceasing to advance the former. Because we have surpluses of certain things does not mean that we have too much wealth or too much power to produce wealth. —Henry A. Wallace, Secretary of Agriculture, in *Science*, Aug. 3, 1934, pp. 105–106.

The Strength of ALLOY-STEEL BOLTS

By W. F. Davidson[1]

In most engineering applications of data on strength of materials as determined by tests on small specimens, more or less uncertainty is involved in computing the strength of full-size members. This arises both from the non-uniformity inherent in large structural cross-sections and from uncertainties as to stress distribution as influenced by external factors. During recent years, testing machines of increasingly large capacity have become available and permit full-scale tests on many types of structures. However, comparatively little work appears to have been done in relating the strength of full-size alloy bolts and studs to the physical characteristics of standard-size test specimens.

The present report is based on an investigation to determine the relations between these factors for the purpose of checking design and with a view to simplifying acceptance-test procedure. The machining of tensile-test specimens from heat-treated alloy steels is a difficult and expensive operation and it was hoped that means could be found to avoid the necessity of such operations.

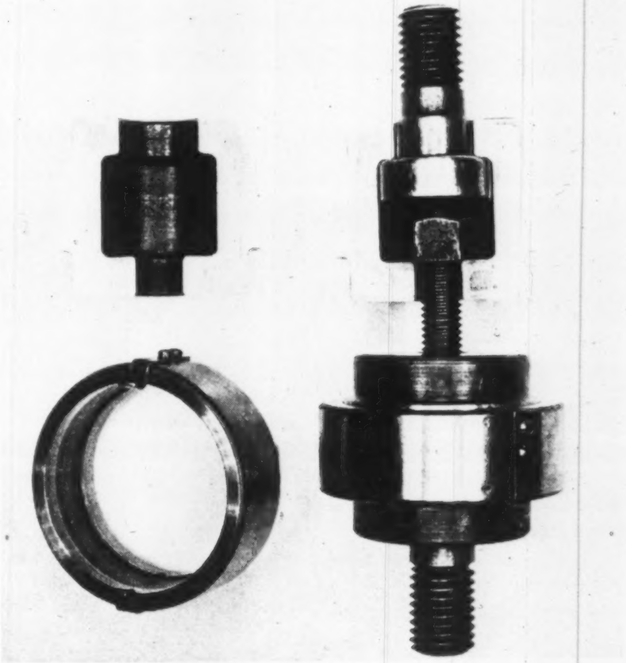

FIG. 1 SPECIAL GRIP FOR TESTING FULL-SIZE STUD

In connection with the routine acceptance tests on about 30 lots of alloy-steel bolting material for high-temperature pipe-line service, 37 pairs of samples were selected at random. Three manufacturers were represented, but the type of steel was generally the same; namely, a chrome-nickel heat-treated alloy. The samples ranged in diameter from $^{1}/_{2}$ in. to 1 in. and in length from 2 in. to $7^{1}/_{4}$ in. One specimen of each pair was turned down to form a standard $^{1}/_{2}$-in. by 2-in. test specimen. (For the $^{1}/_{2}$-in. and $^{5}/_{8}$-in. studs a smaller proportional size was used.) These specimens were tested in a 50,000-lb Amsler hydraulic testing machine with a mechanical multiplier type of extensometer whose least reading was 0.0001 in. The yield points were determined from stress-strain diagrams plotted from load and extensometer readings. The full-size studs were pulled as received, the smaller ones in the Amsler machine and the larger ones in a 300,000-lb Baldwin-Southwark hydraulic testing machine. The rate of strain was kept constant and the yield point determined at the "break" in the stress curve as indicated by a slight drop in the load gage. Most of the studs were pulled using the nuts furnished by the manufacturer and a special type of test grip developed for the purpose. This is shown in Fig. 1. In a few cases the stud was screwed into threaded test grips machined to the thread dimensions of the loosest fitting acceptable nut. All grips were attached to the heads of the testing machine through hemispherical swivel joints.

The comparison of test results obviously can be made on the basis of either unit stresses or total loads. The latter basis is used here as it is of broader general engineering interest.

Table 1 gives the average results for each size of stud. The unit stresses obtained from the tests on the machined specimens are given and from these the expected strength of the full-size stud has been calculated, taking as a cross-section the minimum thread-root area according to American Standard screw-thread dimensions B1A-1924. These figures are compared with the actual test results. It will be noted that, in general, very satisfactory agreement exists. The calculated and measured yield points do not differ by more than 6 per cent and have an average deviation for all

TABLE 1

				Full-size specimens					
				Calculated[a]		Actual		Ratio Actual/calc.	
Stud diameter, in.	No. of specimens	Machined specimens Yield point, lb/in.^{2}	Machined specimens Ultimate strength, lb/in.^{2}	Yield point, lb	Ultimate strength, lb	Yield point, lb	Ultimate strength, lb	Yield point	Ultimate strength
$^{1}/_{2}$	3	122.3	135.0	15,370	16,970	15,000	17,900	0.976	1.055
$^{5}/_{8}$	15	132.5	140.6	26,740	28,370	26,130	30,010	0.977	1.058
$^{3}/_{4}$	4	127.5	139.5	38,500	42,130	39,250	46,050	1.019	1.093
$^{7}/_{8}$	6	121.4	136.2	50,900	57,110	53,220	61,330	1.046	1.074
1	9	117.3	137.9	64,630	75,980	68,220	78,880	1.056	1.038
							Weighted av	1.012	1.059

[a] Calculated on basic root of screw.

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[1] Director of Research, Brooklyn Edison Co., Brooklyn, N. Y. Mem. A.S.M.E.

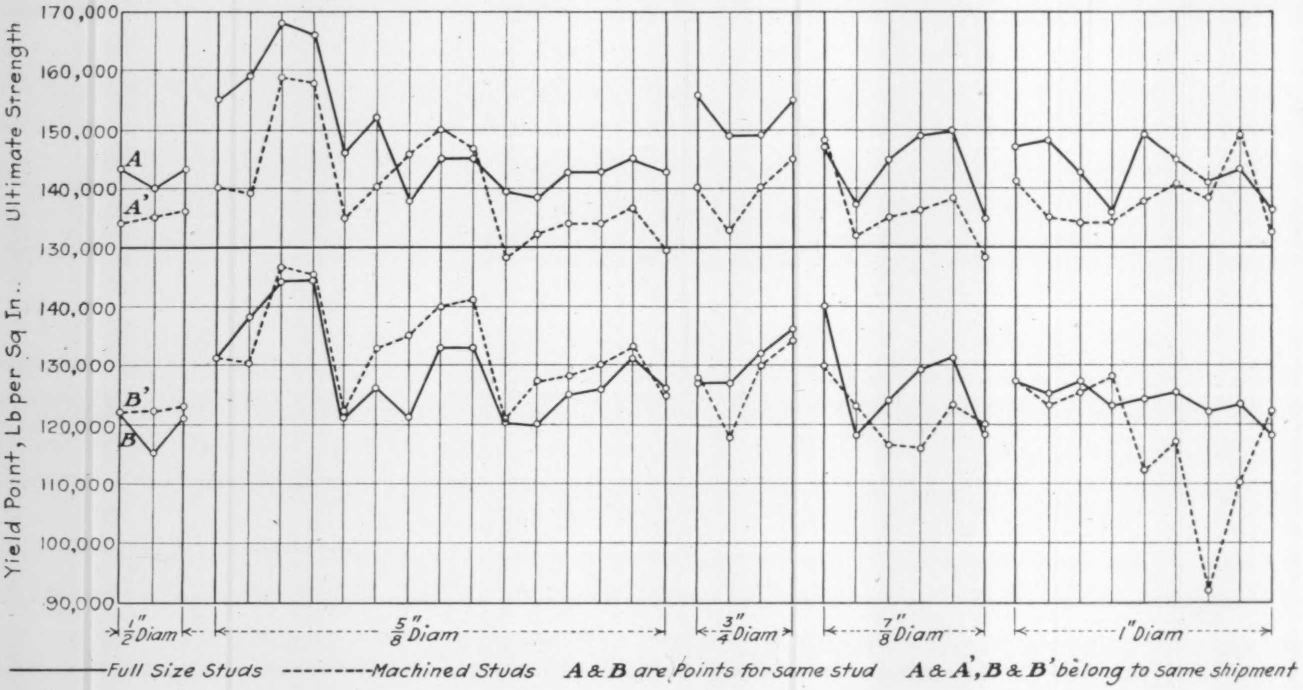


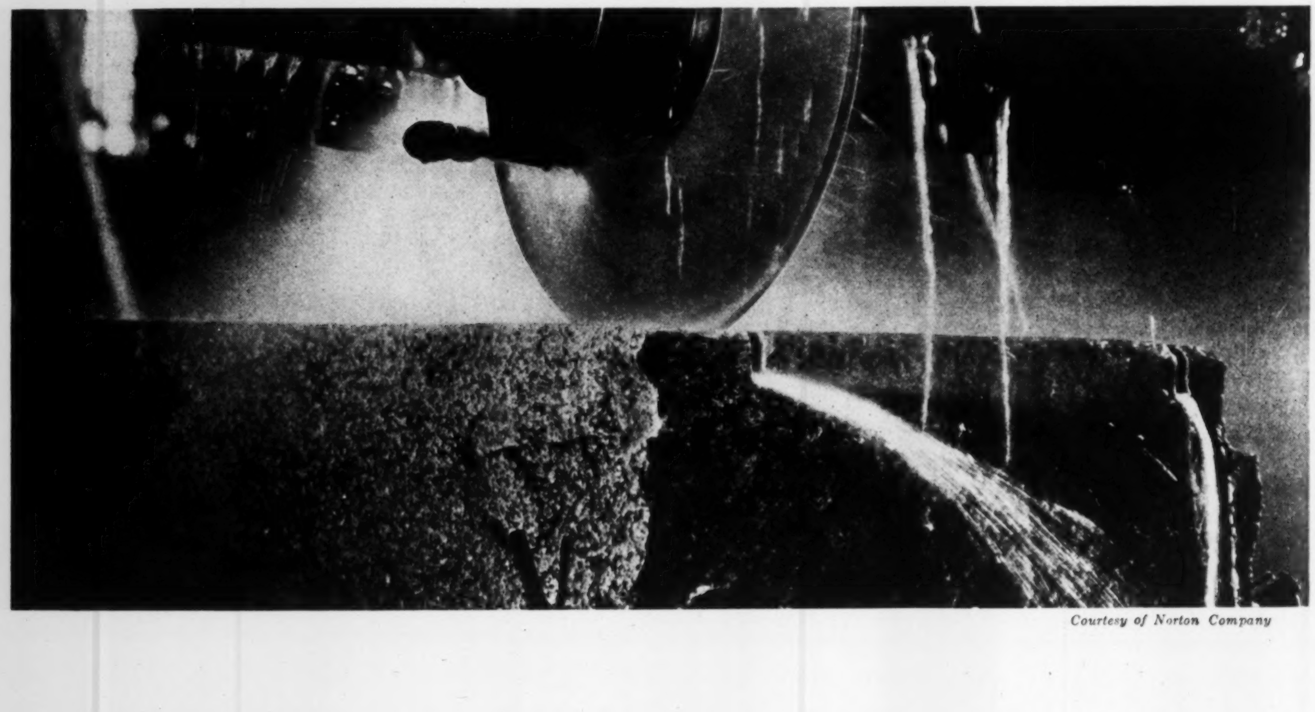
FIG. 2 COMPARISON OF PHYSICAL PROPERTIES OF FULL-SIZE AND MACHINED ALLOY-STEEL STUDS

sizes of only 1 per cent. For the ultimate strength, the deviations are somewhat larger, although the maximum is 9 per cent and this in the direction of giving actual strengths higher than would be obtained from calculations.

Fig. 2 gives a comparison of individual test results. Each point on the graph represents one determination, *B* and *A* being the yield point and ultimate strength, respectively, calculated from the test results on a full-size stud, while *B'* and *A'* are yield point and ultimate stress, respectively, of a machined specimen from the same shipment. Thus the points on each vertical line are on samples from one shipment.

Taken as a whole, the results show that design calculations based on unit stress determined by specimen test and thread-root area will be conservative. From this it follows that if tensile tests on full-size specimens show strength in excess of those calculated from the minimum permissible unit stresses stipulated in the specification, acceptance under the specification may reasonably be made.

A point of further interest is the observation that with all of the full-size specimens, the condition of the threads after completion of the test was such that the nuts or grips could be removed with about the same facility with which they were installed; thus indicating that the threading and nuts are of adequate design.



Courtesy of Norton Company

VOLUME CHANGE *in* PRESSURE VESSELS

By H. B. HOLROYD[1]

THE MOST natural method of procedure in calculating the change in volume of pressure vessels due to deformation of the walls by pressure is to calculate the volumes for the unstressed and the stressed conditions and to subtract to determine the difference. However, another method based on energy may be applied more easily in some cases. This problem is of interest in designing devices for absorbing hydraulic shocks.

Consider that a uniform pressure alone causes deformations of the walls of the vessel. We may consider that this pressure is applied by means of an incompressible fluid having no weight. That such a fluid does not exist is of no importance, since this is only a method of thinking; there is no approximation due to this assumption.

No energy can be stored in an inelastic medium. Hence all of the energy used in building up the pressure must be stored in the walls of the vessel, for no energy is lost and there is no other place for storing it. Thus

$$du = p\,dv \qquad [1]$$

in which u = energy of deformation

p = pressure
v = volume of vessel.

Integrating, we have

$$v = \int_0^p \frac{du}{p} \qquad [2]$$

Equation [2] is valid in all cases. Another form may be derived for the cases in which the law of superposition is applicable. In these cases, the strain energy is a quadratic function of the forces; this means that the generalized force, pressure, and the strain energy are related by the equation

$$u = kp^2 \qquad [3]$$

in which k is a constant. Substituting Equation [3] in Equation [2] and integrating gives

$$v = 2kp = 2u/p \qquad [4]$$

Equation [4] shows that the error made in the change in volume of the vessel due to neglecting some of the strain energy is proportional to the energy which is neglected. The error made by assuming that the principle of superposition is strictly valid when it is only approximately true may be more serious; it cannot be estimated for the general case. Equation [2] should be used when the principle of superposition is inapplicable.

The maximum-strain-energy theory may be applied to give an interesting result. According to this theory, the strain energy per unit volume of a material stressed up to the yield point has a definite value which is independent of the manner in which the stresses are applied. The greatest energy can be stored in a given amount of material when it is uniformly stressed, and hence the change in the volume of a vessel for a given amount of material and a given pressure is greatest when the form of the vessel is such that the stresses are uniform. The change in volume of the vessel must be less than this possible maximum when the stresses are not uniform. From the maximum-strain-energy theory, the maximum stored energy is

$$U_{\max} = \frac{S^2}{2E}q \qquad [5]$$

in which $U_{\max}$ = maximum strain energy
S = stress at the yield point
E = modulus of elasticity
q = volume of the material.

The maximum change in volume of the vessel, under the conditions that the law of superposition is applicable and that the pressure stresses the material uniformly, is found from Equations [4] and [5] to be

$$v_{\max} = \frac{S^2}{Ep}q \qquad [6]$$

No assumptions have been made in regard to the physical shape of the stressed materials. Equation [4] is applicable, for example, to the case in which the pressure vessel is a cylinder with a piston held by a spring; only the spring stores any energy. The material of the spring is not uniformly stressed, and hence the change in volume is less than the possible maximum which could be obtained from the material of the spring used in the form of a sphere. However, this form would be very inconvenient and too expensive in most applications. If the efficiency in the use of the material to store energy is known, the strain-energy method of calculating the change in volume is an easy method for calculating the amount of material.

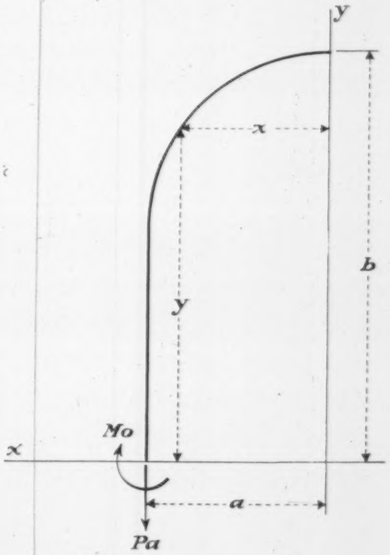


FIG. 1

AN APPLICATION OF THE METHOD

The method may be applied to the approximate calculation of the change in the volume of a vessel with thin walls having a cross-section consisting of two semi-circles joined by straight lines as shown in Fig. 1. (See "Strength of Materials," by

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[1] Altadena, Calif.

Timoshenko, Vol. 2, p. 443.) If the vessel is rather long in respect to the dimensions of the cross-section, the change in volume will be approximately porportional to the length; hence it is sufficient to calculate the change in the volume of the vessel per unit length. If the vessel is quite flat, that is, if the dimension b, is several times the dimension a, nearly all of the energy will be stored by bending the walls rather than by stretching them.

The statically indeterminate moment M_0 may be calculated by the method explained in the reference cited:

$$M = M_0 - pa(a-x) + \frac{p(a-x)^2}{2} + \frac{py^2}{2}$$

$$= M_0 - \frac{pa^2}{2} + \frac{px^2}{2} + \frac{py^2}{2} \ldots\ldots\ldots\ldots\ldots\ldots [7]$$

The total strain energy in one quadrant is

$$u = \frac{1}{2EI_z}\int_0^S (M_0 - pa^2 + px^2 + py^2)^2 ds \ldots\ldots [8]$$

in which I_z is the moment of inertia of the wall of the vessel per unit length about the neutral axis of bending.

Since M_0 does not produce any angular deflection, Castigliano's theorem states that

$$\frac{\partial u}{\partial M_0} = 0 \ldots\ldots\ldots\ldots\ldots\ldots [9]$$

From Equations [8] and [9]

$$M_0 = \frac{pa^2}{2} = \frac{p}{2s}(I_z + I_y) \ldots\ldots\ldots\ldots [10]$$

in which s is the length of one quadrant, or

$$s = b - a + \frac{\pi}{2}a \ldots\ldots\ldots\ldots\ldots\ldots [11]$$

$$I_z = \int_0^S y^2 ds$$

$$I_y = \int_0^S x^2 ds$$

Integrations give for the values of I_z and I_y

$$I_z = \frac{1}{3}(b-a)^3 + \frac{\pi a}{2}(b-a)^2 + \frac{\pi}{4}a^3 + 2a^2(b-a) \ldots [12]$$

$$I_y = a^2(b-a) + \frac{\pi}{4}a^3 \ldots\ldots\ldots\ldots [13]$$

The total energy is four times the energy stored in one quadrant; since we have M_0 from Equation [10], the total energy is

$$u = \frac{p^2}{8EI_z}\int_0^S \left[x^2 + y^2 - \frac{I_z + I_y}{s}\right]^2 ds \ldots\ldots [14]$$

The integration of this equation is long and tedious but perfectly straightforward; the final result is

$$u = \frac{p^2}{2EI_z}\left[\frac{\pi}{2}a^5 + 5a^4(b-a) + 2\pi a^3(b-a)^2 + \frac{14}{3}a^2(b-a)^3 + \frac{\pi}{2}a(b-a)^4 + \frac{1}{5}(b-a)^5 - \frac{(I_z+I_y)^2}{b - a + \frac{\pi}{2}a}\right] \ldots\ldots [15]$$

Equation [15] may be substituted in Equation [4] to give the change in volume per unit length of the vessel.

F. S. Lincoln

DOMESTIC-STOKER PERFORMANCE

Results of Tests of a Domestic Stoker Installed in a Warm-Air Furnace Using Bituminous Coal

By THOMAS G. ESTEP and DAVID C. SAYLOR[1]

SOMETIME ago, at a meeting of a stoker manufacturers' association, the statement was made that domestic stokers could be designed to give heat releases of from 50,000 to 60,000 Btu per cu ft of furnace volume per hr. The authors, being familiar with large stationary power-plant practise where such high heat releases are the exception rather than the rule, wondered if such results were possible in a small domestic heating plant. It was decided, therefore, to make the series of tests reported in this article.

The stoker tested was installed in the usual type of warm-air furnace. Such an installation was selected because it was felt that the minimum heat release would be obtained in a warm-air furnace where the firebox was made up entirely of air-cooled surfaces.

There was nothing unusual about the stoker except that it did provide positive fuel-bed agitation. It had four fixed rates of coal feed. A diagrammatic sketch of the stoker is shown in Fig. 1 and needs no further description.

METHOD OF CONDUCTING THE TESTS

Since the stoker had four rates of coal feed it was decided to make four tests, one for each rate. Enough coal thoroughly mixed was secured for all four tests. It was then sampled and analyzed and the one analysis used for all four tests. The ash-pit refuse from each test was separately sampled and analyzed.

The chimney gases were analyzed every 20 min by means of the standard Orsat apparatus. The carbon-dioxide content was also continuously recorded by a Hays recorder. The temperature of these gases was recorded by a Leeds and Northrup "micromax," using a chromel-alumel thermocouple. Electrical instruments were provided for determining the input to the motor. Inclined-tube manometers of the Ellison type were used for measuring the various draft pressures.

After the fire was started, the stoker feed was set at the desired rate. The combustion air was then adjusted to a minimum for the given rate of feed. It usually required about two hours from the time the fire was started to make the proper adjustments. After conditions had become constant the stoker hopper was filled level with coal and the test started. The tests were run long enough to minimize any error resulting from a variation in the condition of the fire and the level of the coal in the stoker hopper at the beginning and end of the tests.

DISCUSSION OF RESULTS

The principal results of the four tests are presented in Table 1 and in the curves in Fig. 2. It is to be kept in mind that these are tests of a stoker and not a warm-air furnace, so that only the results pertaining to the stoker performance are shown.

The results are, of course, for this particular stoker installed in a warm-air furnace. If the same stoker were installed in a completely water-cooled furnace the results would, no doubt, be different and there is also no doubt but that other stokers will give different results, but it is believed that the data shown are representative of small-stoker performance.

The total power required to operate the stoker and fan varied from 22 kwhr per ton of coal burned at the low rate to 11 kwhr per ton at the high rate. It is usually estimated that the cost of power will be approximately one dollar per ton of coal, hence this figure seems to be conservative. Domestic stokers are usually equipped with a one-quarter-horsepower motor. This size is ample but it should not be any less.

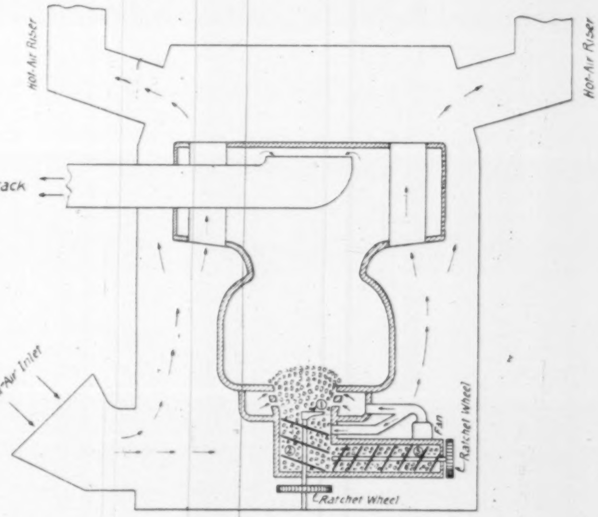

FIG. 1 DIAGRAMMATIC SECTION OF FURNACE AND STOKER
(1 Agitator, 2 and 3 Intermittently operated screws.)

In test No. 4, the draft loss through the fuel bed is a little more than the maximum head produced by this fan, the difference being made up by the draft from the chimney. Hence, test No. 4 is about the highest rate of combustion possible with this fan.

The rate of combustion, pounds of coal per square foot of grate surface per hour, was based on the projected area of the grate, that is, the area of a circle having the diameter equal to the outer ring of air tuyères. The maximum rate of 53.37 lb is about the limit for this stoker. A higher rate of feed would cause the coal to accumulate in the furnace.

The percentage of excess air varied within wide limits and in the usual manner. As already mentioned, before each test was started the stoker was run for some time and the air supply adjusted so that the coal would burn as rapidly as fed. Hence the air used represents about the minimum possible in each test. No carbon monoxide was found in the stack gases in any of the tests but an examination of the curve of excess air shows that some would be expected if the rate of combustion exceeds that of test No. 4.

The total stack loss is high, as is to be expected in an air-cooled furnace. This loss has been included in the test largely to give some idea of the general shape of the curve, because this loss is partly dependent upon the design of the heat-absorbing equipment, the stoker being responsible only for the weight of the gaseous products. Different arrangements of heat-absorbing surfaces will bring about a difference in the stack temperature and hence the stack loss. The curve of this loss, however, should have the same general shape. The stack loss for the highest rate of combustion shows a rapid falling off resulting from the high percentage of carbon dioxide, but this loss will begin to increase beyond this point because of the formation of carbon monoxide.

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[1] Professor and Assistant-Professor, respectively, of Mechanical Engineering, Carnegie Institute of Technology, Pittsburgh, Pa. Members, A.S.M.E.

The ashpit-loss curve is interesting from two angles. First, it is a very small loss, being less than three per cent at the maximum rate of combustion. The rapid increase in this loss beyond test No. 4 is attributable to the fact that the rate of combustion in this test represents the maximum obtainable with the available air supply and any additional coal fed to the furnace will appear unburned in the refuse. The second interesting point is the higher loss at the low rate of combustion than at the two intermediate rates. This higher loss at the low rate of combustion is the result of the large amount of excess air passing through the outer fringe of the fuel bed and cooling the fire at this point until combustion is extinguished. At the rates of combustion for tests No. 2 and No. 3 there is a more uniform dis-

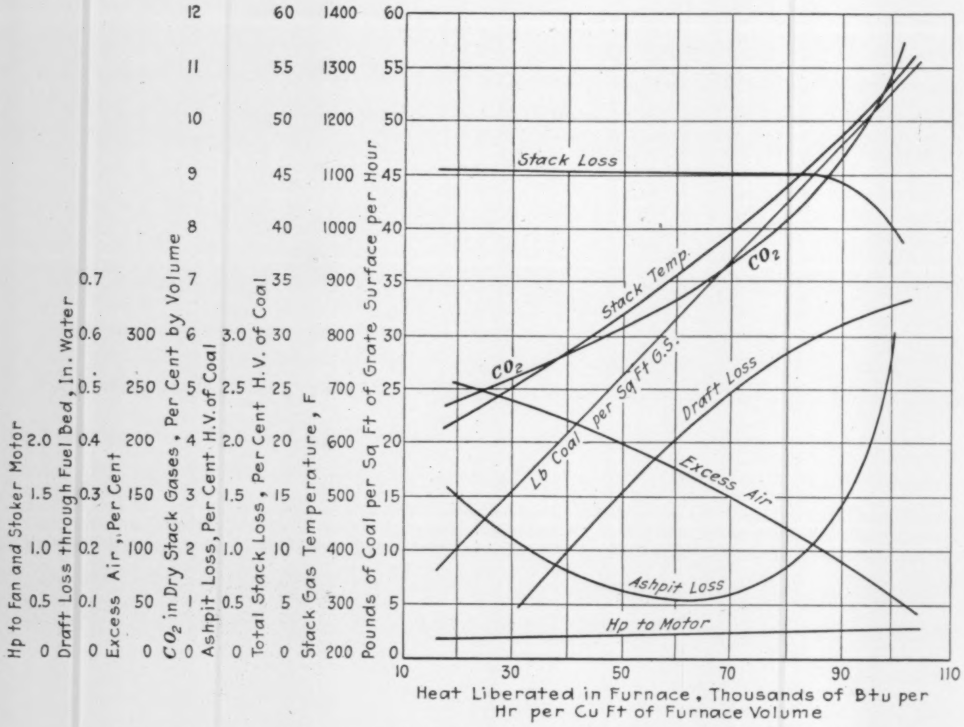


FIG. 2 PERFORMANCE CURVES OF SMALL STOKER INSTALLED IN A WARM-AIR FURNACE USING BITUMINOUS COAL

TABLE 1 PRINCIPAL OBSERVED AND CALCULATED DATA FROM THE TESTS

	Test No. 1	Test No. 2	Test No. 3	Test No. 4
Operation of stoker.....		Continuous		
Notches of coal feed....	1	2	3	4
Attention to fire........	None	None	Removed ashes	Removed ashes
Length of test, hr......	13.5	15.37	12.5	9.0
Furnace draft, in. H_2O..	0.120	0.106	0.110	0.100
Stack draft, in. H_2O....	0.150	0.166	0.190	0.184
Forced draft, in. H_2O...	0.012	0.195	0.500	0.560
Draft loss through fuel bed, in. H_2O.........	0.132	0.301	0.610	0.660
Draft loss through furnace, in. H_2O........	0.030	0.060	0.080	0.084
Power required to operate stoker and fan, hp	0.160	0.206	0.240	0.240
Analysis of stack gases, per cent by volume:				
CO_2................	5.3	6.4	8.8	11.2
O_2.................	14.8	13.5	10.8	8.1
CO.................	0.0	0.0	0.0	0.0
N_2.................	79.9	80.1	80.4	80.7
Temperature of stack gases, F............	734.8	861.0	1129.0	1276.0
Room temperature, F...	80.4	84.8	90.0	87.8
Total weight of coal fired, lb.............	146.3	244.3	344.5	289.0
Coal fired per hour, lb...	10.84	15.90	27.60	32.10
Total weight of ash and refuse, lb...........	13.55	21.91	32.35	32.03
Combustible in refuse, per cent............	9.32	6.33	10.55	24.19
Combustible in refuse, lb	1.26	1.39	3.41	7.75
Combustible in refuse, lb per lb coal...........	0.00861	0.00569	0.00990	0.02681
Loss in refuse, Btu per lb of coal..............	125.7	83.1	144.5	391.4
Loss in refuse, per cent of heating value of coal..	0.94	0.62	1.08	2.92
Heat liberated in furnace, Btu per hr...........	143,893	211,739	365,852	417,576
Furnace volume, cu ft...	4.23	4.23	4.23	4.23
Heat liberated in furnace, Btu per cu ft furnace volume per hr........	34,016	50,055	86,487	98,717
Projected grate area, sq ft	0.6015	0.6015	0.6015	0.6015
Coal per sq ft grate surface per hr, lb........	18.02	261.43	45.89	53.37
Total stack loss, Btu per lb of coal............	6113.0	6111.0	6016.0	5439.0
Total stack loss, per cent of heating value of coal	45.6	45.6	44.9	40.6
Excess air, per cent.....	237.2	180.4	104.2	59.7

Proximate Analysis of Coal

Moisture.........	1.7
Volatile.........	37.9
Fixed carbon.....	52.0
Ash.............	8.4
	100.0

Ultimate Analysis of Coal

Carbon.........	73.20
Hydrogen.......	5.05
Nitrogen.......	1.43
Sulphur........	1.60
Oxygen.........	10.32
Ash............	8.40
	100.00

Heating value of coal as received, 13,400 Btu per lb
Ash-softening temperature, 2450 F.

tribution of the air through the fuel bed and more complete burning of the coal.

While a heat release of nearly 100,000 Btu per cu ft of furnace volume per hour was obtained in test No. 4, this release is too high for this type of furnace. The cast-iron furnace hood warped enough to open some of the joints and the flue-gas baffle plates were badly burned and scaled.

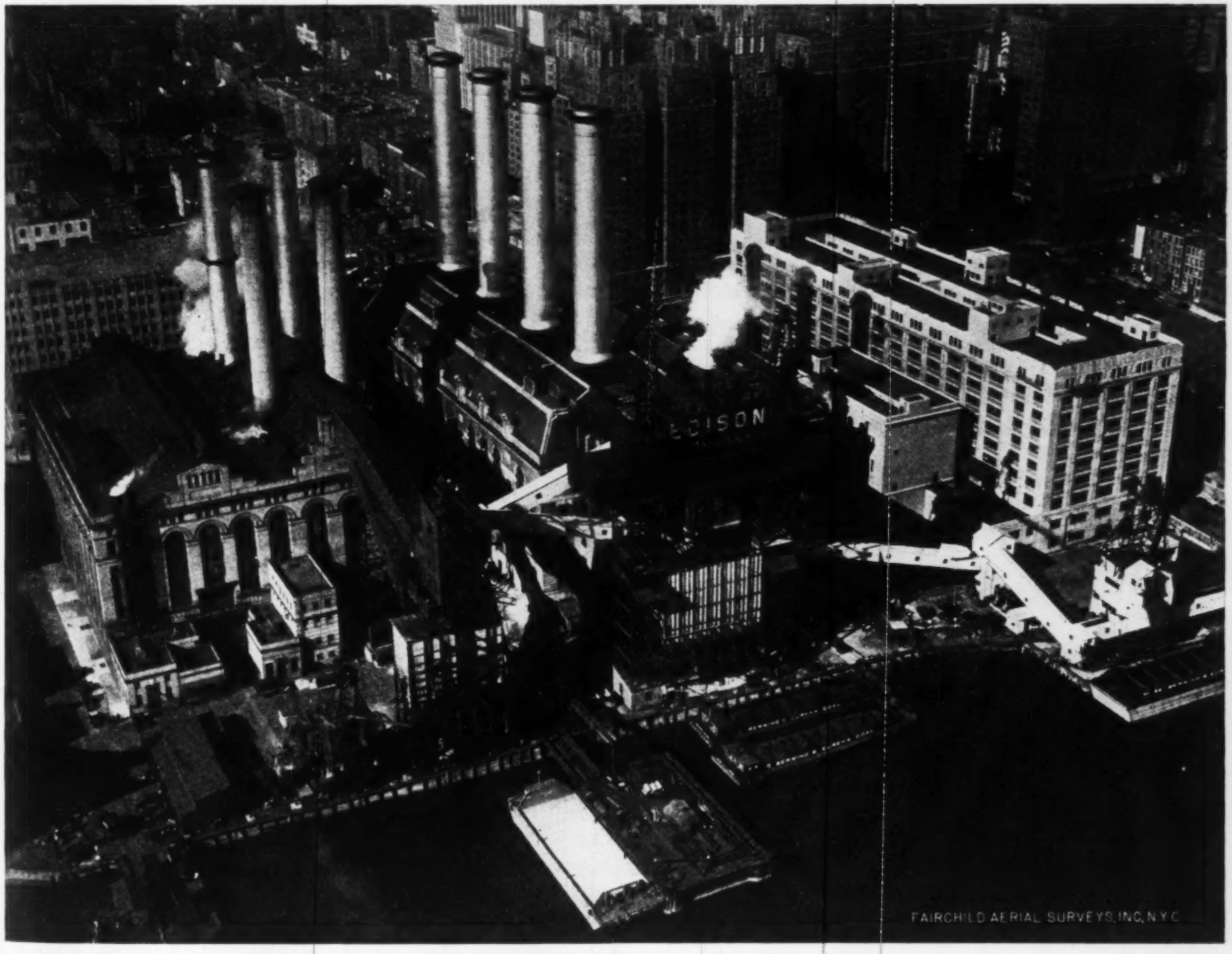
No damage was noted from test No. 3 where a heat release of 86,487 Btu was obtained, but both the stack and riser temperatures were too high for a practical installation. It would seem, therefore, that a design figure of from 50,000 to 60,000 Btu per cu ft of furnace volume per hour could be used but should not be exceeded, particularly in a warm-air heating plant.

It is believed that the higher temperature of the air-cooled furnace is an aid at the lower rates of combustion and a detriment at the higher rates. In other words, it will probably be more difficult to maintain a slow fire in a water-cooled furnace than in an air-cooled furnace, particularly during intermittent operation, but it will be possible to secure a much greater capacity in a smaller space with the former.

The coal used, a washed $^3/_4$ by $1^1/_2$-in. nut, was an ideal fuel for this type of stoker. Its ash-fusion temperature was relatively high (2450 F) and no trouble was experienced from clinker. Although the volatile matter was nearly 38 per cent, no objectionable smoke was produced at any time. The stoker performance was entirely satisfactory in all tests. It is to be particularly noted that no attention was given to the fire after the initial adjustments were made. The authors have made other tests with a very inferior grade of coal and secured equally satisfactory results.

Whether or not the better efficiency secured by stoker firing over hand firing will pay the fixed and operating charges of the stoker will depend primarily on the cost of coal, but the stoker will have the additional advantages of convenience, smokeless combustion of high-volatile coals, and more uniform residence temperatures. It is difficult to place a monetary value on these additional advantages but most home owners will agree that they are of great value.

It is hoped that the publication of these tests and comments will stimulate the development of small stokers for domestic heating units by showing the possibilities of such equipment, that manufacturers will publish similar information on other types and installations, and that prospective purchasers will have more confidence in the performance of such equipment. The small stoker is certainly one way of eliminating the smoke nuisance from domestic heating plants located in the bituminous-coal-consuming districts.



AIRPLANE VIEW OF NEW YORK EDISON COMPANY'S WATERSIDE STATIONS

MECHANICAL
ENGINEERING

Vol. 56 SEPTEMBER, 1934 No. 9

GEORGE A. STETSON, *Editor*

Junior Activities

INSTEAD of the customary frontispiece, an open letter from President Doty to the members of The American Society of Mechanical Engineers will be found opposite the first article of this month's issue of MECHANICAL ENGINEERING. The letter is a report on the status of the Society and a briefly outlined program of fall activities. One of the activities that President Doty lists for immediate development involves the large and increasing number of younger men recently enrolled in society membership. To this group attention has frequently been drawn because of the very real desire that exists to assist in the satisfactory working out of common and individual problems and because of the enthusiasm with which the Juniors making up the group have been looking after their own interests in many localities.

A report by Gideon M. Varga, a member of the Junior group of the Metropolitan Section of the A.S.M.E., on the activities of that group has recently come to hand.

Organized under the leadership of Charles Hescheles, the Metropolitan Juniors laid out a varied program of meetings and inspection trips. Speakers included members of the group as well as outsiders and older engineers. In addition to holding regular meetings, several committees were formed. One of these is engaged in an investigation of the manner in which engineers may be of service in industries and businesses which heretofore have used little or no engineering. Another, formed by a group of sales engineers, has been combining with regular sales work an investigation of employment opportunities. Study groups devoted to physical chemistry, finance, and literature were formed. Moreover, a publicity committee has been at work, enlisting the interest of executives in the unusual opportunity that now exists to secure the services of engineers.

In the conclusion of the report the statement is made that the object of the activities is to make membership in the Society a real asset. The Juniors have discovered that the value of the asset depends upon the extent to which the Junior avails himself of the opportunities of membership to work actively in the common interests of the profession and enlarge his knowledge and friendships as greatly as possible. It has always been the experience of members of professional societies that those who get the most out of the experience put the most into it. The Juniors of the Los Angeles, Metropolitan, and other sections where the younger men are taking an active part are learning this early enough in their careers to benefit hugely from it, now and later. Juniors who wish to take an active part in Society work and older men who are interested in assisting Juniors should get in touch with their local groups or with headquarters.

Men—Not Machines—Are to Blame

A SCAPEGOAT to suffer for the wrongdoings of others has been sought in all ages and in all civilizations so that the people may sin realistically and suffer vicariously. Failing such a convenience, the technique may be reversed, sinning by proxy and thus escaping the inevitable consequences. So it appears to be the case with the drama, and particularly with the movies. Otherwise law-abiding and moral citizens permit themselves to be entertained by tasting vicariously the sweet joys of forbidden experiences at the movies, afterward to wander homeward and resume their customary pursuits in the paths of rectitude without feeling the vengeance of outraged society or the prickings of conscience that punish and disturb the ill-fated characters of the screen. Thus the Casper Milquetoasts for the last 3000 years have felt themselves to be amorous Paris, crafty Ulysses, dauntless Achilles, or outraged Menelaus.

Modern counterparts of the tragic figures of ancient times flit across the screens of countless motion-picture houses as apexes of more modern triangles and as the gangster substitutes of invading Greek warriors. We flock to see them, and the purveyors of this type of entertainment shrewdly learn that we like to imagine ourselves as men of strong emotions and reckless courage, and give us more of it, just as Scheherazade beguiled her lord for one thousand and one nights. Give us more, until they give us too much and certain groups protest, as the great religious groups of today are protesting at crime and sex and filth in movies and magazines and on the stage.

Engineers become identified with this moral controversy because of their association with the groups of society that invent such means of glorifying crime and passion as the movies and other mechanical amusement devices provide. They come in for censure because they have invented these machines, and because they manufacture and maintain them. So Prometheus suffered for stealing the fire of Zeus—so inventors have been accused since the world began, because most inventions are two-edged swords, because most inventors can be used constructively to advance the well-being of mankind, or destructively to hasten degeneration and decay.

The real fault, when such inventions as printing, the movies, and the radio are misused, lies with the upper stratum of intelligent and cultured persons who should direct the progress of the arts toward constructive ends but who are unlikely to recognize the beneficial potentialities of the device until others have prostituted it to ignoble ends. The time for the groups who are now concerned with the effects of the movies on our national character to have become alive to the dangers in the invention was when it was first used. Eternal vigilance to keep the use of inventions for beneficial purposes is always necessary. Engineers, who should have active imaginations when it comes to forecasting what an invention may develop into, should be among the first to point out the hazards and arouse their fellowmen to a desire to direct the development along lines that will bring the greatest benefit and the least harm to them.

Aeronautics Still Young

IT WAS just one hundred years ago that Samuel Pierpont Langley, who devoted so many years to a scientific study of mechanical flight, was born. The bitter disappointment of his life—his failure to fly the airplanes on which he had spent so much time and money—has been forgotten by most persons. He stood on the threshold of a great accomplishment over which others passed to prove that men can fly. Today, no one would call a man who believed in mechanical flight an idle dreamer.

In the short space of time in which this accomplishment of flying has been learned, we have grown to think of the airplane as the only practical form of machine. Then came the autogiro, a strange looking "windmill" that could perform feats of landing and taking-off in a leisurely manner impossible for the airplane. Who knows but what some other device may better this accomplishment and attain the vertical and hovering flight that inventors look for in the helicopter?

Several years ago, at the First National Aeronautics Meeting of the A.S.M.E., Frederick Kurt Kirsten presented an analysis of cycloidal propulsion applied to aircraft. Aircraft with this type of propulsion can hover and rise vertically. More recently Rohrbach, in Germany, and Platt, in the United States, have experimented on airplanes having rotating wings. And in France, C. B. Strandgren is developing an aircraft known as the Strandgren cyclogiro. A description from an article published by the inventor in *L'Aerophile* was abstracted in the November, 1933, issue of MECHANICAL ENGINEERING. With the purpose of informing readers directly about this unusual type of aircraft, M. Strandgren was asked to prepare the article that appears elsewhere in this issue. Regardless of what its future history may be, such a device as the cyclogiro should be brought to the attention of engineers as a reminder that aeronautical engineering is still young.

"An End to Unemployment"

LAST month mention was made on this page of an address by Mr. Flanders on unemployment. The complete text of this address, "An End to Unemployment," appears as the leading article this month.

It would be easy to maintain the position that unemployment is one of the greatest, if not the greatest, problem facing the civilized world at the present time. Whether it be viewed as a result of economic maladjustments or as the cause of human suffering and of political and economic insecurity is a distinction of less importance than the stark realism of the situation which unemployment presents to its victims and to those who are concerned with the well-being of their fellowmen. Hence, any proposal to end unemployment demands consideration; and if the proposal comes from a man known for his sound thinking and wide knowledge of the conditions and influences of industry, society, and government, it deserves particularly careful study.

Mr. Flanders' proposal recognizes the inevitability of conditions that make for unemployment, and against them he would raise three lines of defense. For those fluctuations which are seasonal and cyclical in character, and for the general hazards caused by obsolescence and the decay of industrial institutions, unemployment reserves are urged. For prolonged periods of economic readjustment, such as that through which we are passing, he finds in the CCC, the CWA, and the FERA possible substitutes for the enlistment of services and enthusiasms so generously contributed, in time of need, to the destructive purposes of war, but directed in this instance to the "advance of a multitude of socially desirable projects." Such measures, however, call for the third defense, a permanently and efficiently organized non-political public-works administration. The instrument by which these lines of defense are to function is a national system of employment offices, to which must go "every one who would draw on this unemployment reserve and every one who seeks subsistence employment."

In examining Mr. Flanders' proposals we find that the country is already moving rapidly in the direction of greater economic security in the form of unemployment "insurance."

As to his second line of defense, relief organizations, Federal and local, are providing much experience with the administration of great groups of the unemployed. These organizations are woefully lacking in that Pentecostal zeal which is the "moral equivalent of war," yet, if we are to believe reports from Europe, it is possible to arouse and sustain such enthusiasm. While we would hope to avoid a restless marching up and down of nationally minded young people, and the inefficiency of building subways in one's spare time, we could profit from a coordination of individual purposes in a great and socially desirable objective.

In the case of the third defense, our opinions and objectives with regard to public works are far from being unanimous, and our administration of them has uncovered some of the pitfalls that lie in the way. Assurance that Mr. Flanders' qualifications—efficient, permanent, and non-political—can become accurately descriptive would be welcome if this third line of defense is to be relied upon in the future.

Finally, as Mr. Flanders points out, much will depend on the employment offices and their administration, and much good may be hoped to flow from them. Today the controversy between the American Federation of Labor and the Chamber of Commerce of the United States over the accuracy of our figures on unemployment emphasizes the fact that we have no more than estimates to go by in trying to arrive at unemployment facts.

SURVEY OF ENGINEERING PROGRESS

A Review of Attainment in Mechanical Engineering and Related Fields

STEAM TURBINES FOR VARIABLE PRESSURES

IF WE consider several turbines having the same clearances and the same sums of squares of peripheral velocities, it will be found that for the same adiabatic heads the efficiency is a function exclusively of the amount of steam flowing through the turbine. It is immaterial whether the turbine is of the impulse or reaction type, whether the flow is axial or radial. Under these assumptions the efficiency depends exclusively on the steam volume ξ. The recognition of this fact led to the development of the "sliding (variable) pressure" process ("precompression regulation") for steam turbines. In this process the temperature of the steam ahead of the turbine is maintained constant, while the pressure, and hence the amount of steam, is varied in accordance with the load. The turbine is so proportioned that it operates with its best efficiency at the load occurring most frequently. When a greater or lesser overload occurs the steam pressure is varied accordingly. The surface condenser is dimensioned to handle the amount of steam corresponding to the most favorable conditions of operation of the turbines.

For greater outputs and hence greater masses of steam handled D (kilograms per hour), the steam pressure p_2 in the condenser follows a law which may be generally expressed in the form

$$p_2 = aD^2 + b \qquad [1]$$

where a and b are constants. It further appears that the exhaust steam Dv (cubic meters per hour) remains constant for a large range of quantities of steam passed through the turbine. In this expression v is the specific volume of steam in cubic meters per kilgram. As the amount of steam passed through increases, v becomes smaller, so that the exit losses with a given exit area are a minimum for the normal load and decrease with increasing load.

VARIATION OF EFFICIENCY

If we assume for the sake of simplicity that the amount of exhaust steam remains constant, which applies also to the exit velocity and exit losses, it can be shown that the adiabatic drop for various initial pressures is practically constant. As we are dealing here primarily with turbines driving alternating-current generators which have to run at constant speed, it will appear that under the conditions specified the quality factor of the turbine will also be constant.

It would appear, therefore, that the efficiency of the turbine can be determined exclusively from the effect of the influence of the volume factor ξ. To illuminate these relations the author uses the well-known functional dependency of the efficiency η on the volume factor, which he expresses as follows:

$$\eta = \eta_0 - b\xi \qquad [2]$$

where

$$\xi = \frac{P_1 - P_2}{DH_0} \qquad [3]$$

and P_1 is the initial pressure, P_2 the final pressure, D the amount of steam passing in kilograms per hour, H_0 the adiabatic heat drop, η_0 the theoretical efficiency of a steam turbine passing an infinitely large quantity of steam. The magnitude of b varies within narrow limits as a function of the volume factor. For a large range of values, however, it can be assumed to be constant. This appears clearly from a curve in the original article showing the relationship between the efficiency and the volume factor.

As the final pressure P_2 (vacuum) is small as compared with the initial pressure P_1, the volume factor ξ can be expressed as follows in a simplified manner:

$$\xi = \frac{P_1}{DH_0} = \frac{\text{constant}}{H_0} \qquad [4]$$

which is correct, as, in view of what has been stated, the mass of steam D flowing through the turbine may be considered as being proportional to the initial pressure.

The efficiency, therefore, may ultimately be expressed in a formula in which only constant numbers occur. Under these conditions the influence of the volume factor on the efficiency with the process here referred to disappears, and we come to the important conclusion that over a large range of loads the efficiency of the turbine set remains practically constant.

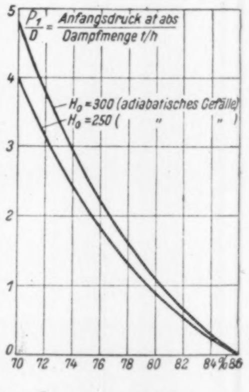


FIG. 1 RELATIONSHIP BETWEEN EFFICIENCY AND PRESSURE AT DIFFERENT ADIABATIC HEADS

(*Anfansdruck at abs/Dampfmenge t/h* = Initial pressure in atmospheres absolute/quantity of steam in tons per hr; *adiabatisches Gefalle* = adiabatic head; *Wirkungsgrad* = efficiency.)

From the schedule of operation of a power plant the normal load, which means the load that occurs most frequently, may be calculated. It will be a fraction of the maximum load, and the most economical pressure for this normal load must be determined in accordance with the amount of steam flowing at that load through the turbine. The curves in Fig. 1 may be used for determining this most economical "normal" pressure. These curves have been developed for condensing turbines and show that the efficiency of a steam turbine is determined by the value of the ratio of the pressure to the mass of steam flowing. The adiabatic drop is of only minor importance. The procedure is as follows: First, plot on the enthalpy-entropy diagram the lower pressure limit determined by the temperature of the cooling water available and then the quantity of steam based on an assumed steam consumption for the normal load. The approximate adiabatic head is then determined.

The initial pressure and the initial temperature (for the latter about 450 C may be set as the upper limit) are also determined by the terminal moisture content in the last stage of the turbine; this should amount to from 10 to 12 per cent. This limitation is necessary in order to prevent particularly active erosion in the last low-pressure stages of the turbine occurring at the most frequently encountered load. If the conditions are so represented, the efficiency can be determined from Fig. 2, and by proper justification the values can be so improved as to obtain sufficiently reliable data for a general calculation.

The author proceeds next to discuss the difference between cut-off and variable-pressure regulation. The output L in kilogram-calories per hour, or N in kilowatts, is given by the following formula:

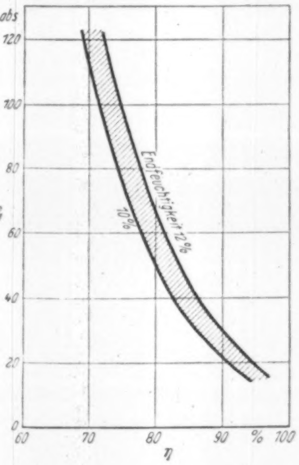


FIG. 2 EFFICIENCIES AT VARIOUS INITIAL PRESSURES P_1 FOR FINAL MOISTURE OF 10 TO 12 PER CENT

$$L = 860\ N = D\eta H_0 \ldots\ldots\ldots\ldots\ldots [5]$$

which, taken jointly with Equations [2] and [4], gives

$$L = D\eta_0 H_0 - bP_1 \ldots\ldots\ldots\ldots\ldots [6]$$

and from this

$$D = \frac{L}{\eta_0 H_0} + \frac{b}{\eta_0 H_0} P_1 = \alpha N + \beta \ldots\ldots\ldots [7]$$

where α and β are constants.

In the case of an ideal cut-off regulation (P_1 = constant) there will be a straight-line relationship between D and N, Fig. 3. In Equation [7] β represents the steam consumption at no load and therefore can be expressed as follows:

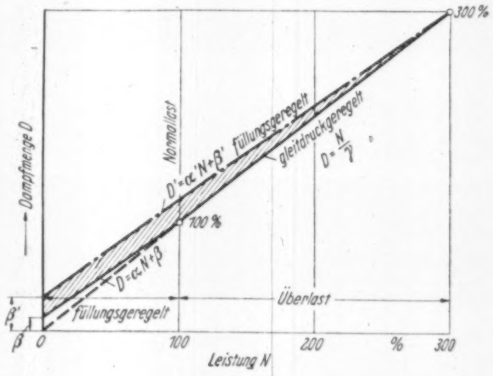


FIG. 3 COMPARISON BETWEEN CUT-OFF AND VARIABLE-PRESSURE GOVERNING, 200 PER CENT OVERLOAD

(*Dampfmenge* = quantity of steam; *Normallast* = normal load; *Füllungsgeregelt* = cut-off governed; *Gleitdruckgeregelt* = variable-pressure governed; *Überlast* = overload; *Leistung* = output.)

$$\beta = \frac{bP_1}{\eta_0 H_0} \ldots\ldots\ldots\ldots\ldots [8]$$

The consumption at no load is therefore in the main dependent on the steam pressure P_1.

On the other hand, in the variable-pressure regulation, the amount of steam D is proportional to the initial pressure, and, hence, $D = cP_1$, where c is a constant. This, taken in conjunction with Equation [7], leads to the following expression of functional relationship

$$D = \frac{L}{\eta_0 H_0} + \frac{bD}{c\eta_0 H_0} = \frac{N}{\gamma} \ldots\ldots\ldots\ldots [9]$$

It would appear, therefore, that a variable-pressure plant follows up to the normal load the rule of Equation [7] and thereafter that of Equation [9]. If we assume a plant operating under the most favorable conditions with constant pressure, it will be found as shown in Equation [5] that, since such a plant is of larger dimensions, it will have a larger no-load steam consumption, but a decreasing additional steam consumption as it approaches the maximum load.

Moreover, since in the case of a variable-pressure plant Equation [8] holds good, the load can be determined from the pressure P_1.

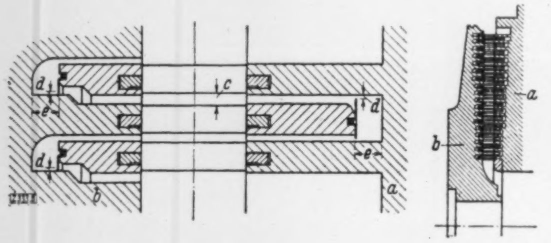
INFLUENCE OF THE USE OF VARIABLE PRESSURE ON THE DESIGN OF STEAM TURBINES

It is only natural that the application of the variable-pressure process should affect the design of the turbine. Because of the closer relationship between the boiler and the steam turbine, the ordinary complicated control provisions used in conventional steam turbines are eliminated, and instead of nozzle regulation, throttling regulation is used up to normal load. At overloads it serves for purposes of fine regulation. High temperatures, which at all loads unfavorably affect the blading, require particular shaping of the rotor. In general, the fundamental principle of the present development of turbine design is evidenced here in the effort to make the running parts of the turbine as small as possible. More emphasis is laid on the exchange of heat between the rotating and stationary parts than ever before, because the efficiency sought is obtained not only by an increase in the number of stages but also by the reduction of leakage losses, which means the decrease of clearances.

It then became necessary, therefore, to develop designs which would satisfy both requirements without affecting the certainty of operation. The purpose of the development is always the production of high-grade counter-pressure turbines working a given amount of steam from a given boiler pressure to the counter pressure. The long rotors consisting of several disks, one behind the other, in which the steam travels axially in the direction of the shaft cannot be used for the highest pressure turbines, and the variable-pressure regulation machines are all of that kind. It becomes necessary to pass to radial steam flow, as an example of which the author shows a Siemens-Schuckert simple radial turbine and a Ljungström counterflow radial turbine. By a proper selection of the angles and diameters, the blading may be so designed as to have blades of practically equal length over a large range of the expansion, thus reducing the hydraulic losses. By heat storage, high degrees of heating may be kept away from the blading bearings, which are of the disk type, so that material elongation cannot occur at all. Because of this, the packing becomes simple and is executed by rolling-in thin nickel sheets at the head end of the blade ring. These sheets have a small clearance and provide backing for the rotor against the housing (see Figs. 4 and 5).

Because of the comparatively short axial length of the rotor, the expansions are small. In turn, because of this, it becomes possible to work with high temperatures and pressures without affecting the reliability of operation. This results in small losses and satisfactory efficiencies, as shown in Fig. 6. The efficiency of 76 per cent at full load (1050 kw) is extraordinarily high for a unit with a steam output of some 13 tons per hour.

As a variable-pressure plant works always above the normal load with the pressure determined by the amount of steam going through, no control apparatus is necessary for that range of loads. Live steam flows into the middle of a radial-flow-type blading disk and expands in one plane to an intermediate pressure which forms the inlet pressure for the next low-pressure section. Similar constructions have been used already to advantage in plants working with constant initial pressure. In such cases the low-pressure part is usually built as an axial-flow bladed drum in a separate housing, or is provided with a pre-set control wheel, if the operation is such that it is necessary to take heating or industrial steam at a certain pressure. Such an installation is shown in the original article. The construction is then similar to that of a bleeder turbine equipped with radial blading in the high-pressure part and axial blading in the low-pressure part. Without condensation, this unit becomes a radial counter-pressure turbine type. Such units are popular in chemical, paper, and sugar plants, where they are giving excellent service. (H. Melan, in *Zeitschrift des Vereines deutscher Ingenieure*, vol. 78, no. 13, March 31, 1934, pp. 402–404, 12 figs., *tcA*)



FIGS. 4 AND 5 CLEARANCE PACKING OF THE SIEMENS-SCHUCKERT RADIAL TURBINE

(*a* = stationary part of the housing with guide blades; *b* = rotor with rotor blades; *c* = radial blade clearance; *d* = radial packing clearance; *e* = axial clearance of the rotor.)

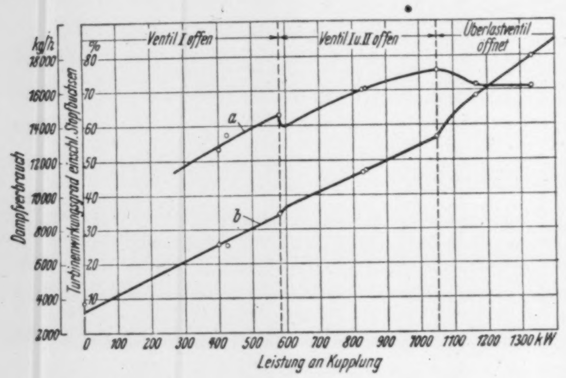


FIG. 6 STEAM CONSUMPTION AND EFFICIENCY OF A RADIAL TURBINE, 3000 RPM, 14 ATM ABS, 350 C, 3 ATM ABS COUNTERPRESSURE

(*a* = steam consumption; *b* = efficiency; abscissas = output at the coupling; ordinates (extreme left) = steam consumption in kg per hr; ordinates (inside row) = turbine efficiency including stuffing boxes; *Ventil I offen* = valve 1 open; *Ventil I und II offen* = valves 1 and 2 open; *Uberlastventil offnet* = overload valve opens.)

Short Abstracts of the Month

AUTOMOTIVE ENGINEERING

Gas From Wood as a Fuel for Motor Trucks

FOR purposes of tests a Vomag truck built in 1925 was reconstructed. The truck weighs 5.1 tons and can carry a useful load of 3 tons. It had a four-cylinder motor developing about 55 hp at 1050 rpm on gasoline and working with a compression ratio of 4.3 to 1. In the present machine the gas producer is suspended in a frame attached to the rear of the platform. This was done in order to permit quickly changing the various types and sizes of gas producers that were used. It is intended that in commercial units the gas producer will be located immediately behind or next to the driver's cab, as the suspension of the producer at the end of the platform interferes with the loading and unloading of the truck. Before going into the motor, the gas passes through five cleaners. By the installation of new pistons the compression ratio of the motor was changed to 9 to 1. It has been found that the most favorable timing of the ignition is at a crank angle 38 deg ahead of the upper dead center. The tests, covering fuel consumptions, acceleration, general behavior of the gas producer and of the vehicle, are described in the original article. On the whole, the tests extended over a distance of about 5000 km and were carried on with four different kinds of gas producers. From the tests it would appear that gas from wood may be seriously considered as a fuel for operating trucks and buses over long distances. Further development of wood-gas applications in power transportation will essentially be dependent upon changes in gasoline and wood prices, and also on the extent to which the future manufacturers of gas producers will succeed in designing them so as to increase their reliability of operation and life, as well as simplify their operation. The engine does not deliver the same power on wood gas as on gasoline and its acceleration is apparently very much poorer. This does not matter very much for long-distance traffic, particularly in open country, but makes the truck less adaptable to city-traffic conditions. (W. Bogner, L. Huber, and W. Kamm, in *Zeitschrift des Vereines deutscher Ingenieure*, vol. 78, no. 10, March 10, 1934, pp. 319–324, 20 figs., *p*)

The Mechanical Horse

THE Karrier mechanical horse is a three-wheel tractor, of interest because of the questions of stability of this type of vehicle involved in the design of the machine here described. It is claimed that, generally speaking, it is a fallacy to consider vehicle stability as being dependent upon the number of road wheels only. The other factors involved are (*a*) relative disposition of the wheels, (*b*) disposition and magnitude of component weight at the front end, (*c*) arrangement of chassis components about the longitudinal axis, (*d*) the method of support and dimensions of the front suspension. The main feature of interest in the Karrier mechanical horse is the use of a single wheel at the front with its unorthodox suspension. The side-members are carried upward at the fore part of the frame where they are joined to a massive bracing plate which carries at its center the suspension unit, to which is bolted the steering head. (*The Automobile Engineer*, vol. 24, no. 317, March, 1934, pp. 83–90, illustrated, *d*)

LUBRICATION

Lubrication of Explosion-Type Motors With Olive Oil

THE importance of developing lubrication of internal-combustion engines of the explosion type by olive oil lies in the fact that this would provide an outlet for oil which is below the grades that are fit for human consumption. The author reports extensive tests on lubrication of engines by this kind of oil carried out at the Agricultural Engineering Station in Algeria on two motors; one, 100 mm stroke, 65 mm bore, running at 1000 rpm; and the other, 100 mm bore, 160 mm stroke, 1200 rpm, tractor type. The author gives several temperature-viscosity curves, both on new oil and oil that has been in service.

On the whole it has been found that the stability of olive oil is about the same as that of mineral oil or lower. The tests have shown that if the oil has been used for 300 hr without draining, a considerable amount of deposits forms. However, 300 hr is an excessive period of operation between drainings. Otherwise, it seems that olive oil can be used as a substitute for mineral oil, providing certain precautions are adhered to. Among these are the following: (1) The oil should contain no sulphur; (2) its acidity should not exceed 0.6, and in any event 0.8, deg oleic acid; (3) the oil should be drained after every 2000 to 2500 km for automobiles and every 40 hr for tractors; the oil should be drained when hot and the motor should be washed out with new oil for half an hour; (4) the oil-pump filter should be cleaned after every 500 km for automobiles and every day for tractors. It is claimed that if these precautions are taken the use of olive oil produces as against mineral oil an economy of about 65 per cent in lubrication and an increase of power that may exceed 10 per cent. There will not be any noticeable increase in power, however, if a low-viscosity mineral oil has been previously used. Where the use of olive oil produces rapid gumming, it should be discontinued. (A. Bestet, Professor of Agricultural Engineering at the Agricultural Institute of Algeria and Director of the Agricultural Engineering Station, in *Bulletin de la Société d'Encouragement pour L'Industrie Nationale*, vol. 133, no. 3, March, 1934, pp. 193–213, 12 figs., *epA*)

MACHINE DESIGN

Speed Variation by Flexible Hydraulic Transmissions

THIS article describes a number of types of flexible hydraulic transmissions used in industrial plants, principally of types in which the efficiency of maintaining the required variations of speed is of paramount value and the efficiency in the transformation of energy is secondary.

In one of these installations a combination of several units is used to secure synchronous rotary movement. This refers to a two-roll variable-speed sectional drive in a paper machine. The two suction couch rolls must be driven at speeds always proportional to the first press roll and to all of the other rolls on the machine.

Power for the first press roll is delivered by a jackshaft through bevel gearing and a friction clutch from the main line-shaft which drives all of the other sections of the machine. Three pumps and three hydraulic motors are used to drive the two suction rolls with the initial power taken from the jackshaft. All pumps and motors are of the same size and the controls of two of the pumps are connected together so that the strokes of these pumps are changed simultaneously.

Speed of the couch sections of the machine is varied in direct relation to the speed of other sections by varying the speed of the main drive shaft of the machine, which drives the pumping units. The speed of each roll of the couch section is adjusted independently by means of a handwheel control on the pump. This adjustment is so delicate that the rolls can be exactly synchronized or the speed can be adjusted so that the amount of tension in the felt can be regulated by infinite steps.

By making small adjustments in the delivery of the main pumping unit, the load on the felt can be distributed so that each roll takes its proportionate share, or can be so adjusted that one roll takes a much greater share of the load than the other. The pressure gage shows immediately the effect of any adjustment in the speed of the driving motor. Once the speed or relationship of speed between the two rolls is adjusted, the speed relationship will remain constant unless some change is made in the adjustment of the pumping unit which controls the drive.

Next is described the application of a hydraulic variable-speed transmission in copper-cable manufacturing plants where stranding or cabling machines are used. In the ordinary type of drive for this application various sets of change gears are used to vary the speed of the capstan. This means an enormous number of sets of gears with perhaps a one-tooth variation in order to take care of the lays desired. The hydraulic transmission provides any speed desired of the capstan from zero to a maximum. Moreover, in an ordinary cabling machine the spider may be rotated in either direction, depending on the type of rope desired, while the capstan must always rotate in the same direction. With the change-gear drive a reversing clutch is required between the capstan and the spider. The hydraulic transmission takes care of the reverse, as the pump unit can be driven in either direction.

An interesting application of hydraulic variable-speed transmission is embodied in a tension drum employed to stretch the rope while it is being made. A somewhat similar apparatus is used with a constant-tension winder, so arranged that the pull exerted on the material being wound on reels or rolls is constant, regardless of the reel or roll diameter. In this case the transmission equipment consists of a variable-displacement pump having an automatic constant-pressure control and a variable-displacement motor having a similar type of control. The material being wound comes to the reel at a constant rate of linear speed. As the material winds up on the reel, greater torque is required to maintain a constant tension and slower speed is required. The automatic pressure control on the motor automatically increases its displacement to compensate for the increased torque required and thereby reduces its speed a corresponding amount. (Allen F. Clark, in *Machine Design*, vol. 6, no. 1, January, 1934, pp. 20–23, 7 figs., *d*)

MACHINE TOOLS

Drills for Cutting Clean-Edged Holes in Sheet Metal

THE author recommends grinding a drill to the shape that is shown in Fig. 7 with the lip undercut as at *A*. The undercut *A* can be ground with a sharp-cornered wheel. With a drill sharpened in this manner holes can easily be spaced within 0.003 in. of the specified dimensions. The point of the drill must enter the center-punch mark in the work before the outer lips begin to cut. The lips of the drill should be thinned at the point and the center-punch mark should be made light. Care must be taken to see that the drill runs true. (Daniel L. Mather, in *Machinery* (New York), vol. 40, no. 7, March, 1934, p. 399, 1 fig., *d*)

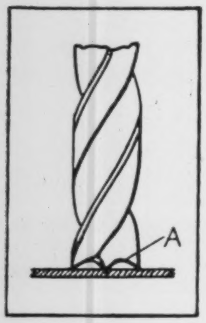


FIG. 7

In commenting on the article quoted, Hammond Mathews, District Supt., Western Colorado Power Co. (*Machinery*, vol. 40, no. 10, June, 1934, p. 584) says that he has employed such a drill for several years for drilling holes from $^{1}/_{2}$ in. up to $1^{1}/_{2}$ in. in copper bus bars used on switchboards and bus members in thicknesses of $^{1}/_{16}$ to $^{3}/_{8}$ in. Previous to learning of this type of drill point it had not been possible to drill copper bus bars with any speed or accuracy. The drill produces a slight burr on the bottom side of the hole which can be trimmed off easily; in fact, on a hurry-up job on brass or copper, a jack-knife will do it neatly.

METALLURGY (See Special Processes: Manufacture of Nickel Parts by Sintering)

POWER-PLANT ENGINEERING

Rapid Valve-Closing Tests on Steam Turbines

THIS article deals primarily with the experience of the Bewag (Berlin Electricity) Company. The author points out that since the rapid valve-closing provision is the last reliance in case of dangerous overspeeding of the turbine, it is particularly important that it should be in good operating condition at all times. It is therefore recommended that it should be tested from time to time. The present rule in the company whose experiences are reported here is to set the speed at which the device comes into action about 10 per cent above the operating speed, and in the case of turbines working at different operating speeds, 10 per cent above the maximum permissible operating speed. This overspeeding, however, produces a tremendous increase in centrifugal-force stresses, which makes the test quite difficult. The author discusses the details of the test as well as of the rapid-closing valves and emphasizes that it is particularly important in overspeed tests to have a short closing period of the rapid-closing type. (Fr. Gropp, in *Elektrizitätswirtschaft*, vol. 33, no. 9, May 15, 1934, pp. 171–173, *dp*)

Loeffler Boiler Tests

THIS article deals with tests and operating results of one of the 132,300-lb boilers at the Caroline Power Station of the Vitkovice Mines Steel and Iron Works in Czechoslovakia.

In the installation in which the tests were carried out, the pump (shown in Fig. 4 in the original article) is driven by a back-pressure turbine supplied with steam from a low-pressure network at about 185 lb per sq in. The packing of the labyrinth gland of the pump is supplied with superheated steam taken from the Loeffler boiler at a point behind the radiant superheater. This steam expands in the labyrinth gland down to the pressure of the turbine steam with which part of it is mixed. The remainder, which expands to about atmospheric pressure, is utilized, together with the exhaust steam from the turbine, for heating the feedwater. The boiler which was tested was fired with a non-gassing pulverized coal prepared from refuse slack. The burners of the Vitkovice type were set in the top of the combustion chamber.

The details of the test methods and apparatus are given in the original article. Tests at partial and variable loads were carried out, including tests on the ability of the plant to accommodate itself to sudden alterations in load. In the latter case the output was raised from 77,200 lb of steam per hr to 132,250 lb in 3 min. To deal with this condition, the pulverized-fuel supply was increased and the revolutions of the pump increased from 4800 to 6700 rpm. The temperature of the steam produced at the lower load was 932 F, and it was found that this temperature could be absolutely maintained during and after the rise in load. In the same way, with a sudden fall in output from 110,000 to 97,000 lb of steam per hour, by reducing the firing and cutting down the revolutions of the pump from 6200 to 4900 rpm, the temperature of the steam produced was maintained at 932 F. These tests have shown that the Loeffler boiler will, at different loads, supply steam at the uniform high superheat of 932 F and that this temperature is maintained even during sudden load variations.

Two further tests showed that it is possible to carry short peak-load demands on the boiler without any adjustment of the firing, simply by altering the speed of the pump.

Additional tests established that the steam leaving the evaporating drums was dry. This is explained by the evaporating arrangement peculiar to the Loeffler boiler in which the rising steam is distributed uniformly over the whole of the evaporating surfaces. Owing to the great height of the steam space, dry steam carrying only very minute or no impurities may be obtained from water containing a relatively high proportion of salts in solution. It was found that the Loeffler boiler tested worked satisfactorily with the water concentration of 3.5 deg Baumé. After 200 hr of operation some of the superheater tubes were cut open and found to be entirely free from deposits. (Prof. E. Josse, Technical University, Berlin, in *Engineering*, vol. 137, no. 3572, June 29, 1934, pp. 722–725, 4 figs., *e*)

Bending of Turbine Rotors and Its Causes

THE author points out the constant tendency toward increase of the number of rotors in modern turbines. Formerly, the turbine handled its steam in seven stages, but in the search for improved efficiency the modern turbine often employs as many as forty. Had an attempt been made to set these forty rotors on a shaft in the same manner as was done with seven stages, the length of the unit would have become prohibitive. Because of this rotors made in a single piece have been introduced. In fact, the shaft and rotor have been made into an indivisible unit. This construction, which was rather unfortunate from the metallurgical point of view, led at first to considerable trouble, in that the rotors easily became bent. Even though modern manufacturers of steam turbines have conquered this difficulty, instances of the bending of rotors still occur from time to time. As the causes of such an occurrence are variable, it was necessary to carry out a systematic investigation in order to find a reason for the occurrence of bending of turbine rotors.

This investigation was undertaken by the Allianz and Stuttgarter Co., a German insurance company, the publishers of the paper here abstracted.

It is pointed out that nickel or chrome-nickel steel are the principal materials for the manufacture of turbine rotors and give some brief information as to the heat treatment of this material which does not radically differ from that used in the United States.

In modern installations the generation of heat stresses in the course of heat treatment is supposed to be eliminated by additions to the steel of either tungsten or molybdenum. Such an alloy after tempering can be slowly cooled in the air without creating temper-porosity. It is claimed that this makes it possible to eliminate to a very large extent stresses in forged pieces. In general, tungsten and molybdenum steel do not have to be reheated after tempering. In particular cases, however, such a reheating may be necessary, especially where it is desired to eliminate the stresses introduced in the material in machining.

In the present case three rotors were the subject of observation and the author points out in general that, in an effort to obtain a favorable rate of steam consumption, turbines were at first built with excessively small and insufficient clearances between the rotating and the stationary parts, which, in certain cases, produced bending of the rotors within as little as a few hours of operation. This defect was obviated by increasing the clearances. Furthermore, in order to avoid the bending of rotors while the turbine was standing still, due to uneven distribution of temperatures in the housing, means have been provided to impart slow rotation to the rotors after the steam has been cut off. Heating turbines while not operating is now generally considered a false policy and is omitted. Of late, however, turbines have been heated while being slowly rotated.

If, however, the rotor has been rusted while standing and the rust, as often happens, is located on one side only, for example, the part which is at the bottom while the turbine is standing still, then even if preheating with simultaneously slow rotation was resorted to, there would be no certainty of uniform heating and hence perfect operation.

If, notwithstanding all precautions, the rotor blades touch, the local rise of temperature brought about thereby will produce stresses in the rotor and these will find expression in the bending of the rotor. There have been, however, cases where bending of the rotor took place without any apparent cause and it is with these cases that the present investigation particularly deals.

The bent rotors were subjected to metallographic and X-ray examinations carried out independently of each other, and it is of interest to note that in all three cases the conclusions derived from either of the lines of investigations concurred with the other. The experimental methods followed are given in detail in the original article.

The metallurgical investigation showed that in rotors I and II there was a considerable difference in structure at the periphery and between the periphery and the inner parts of the rotor. On the other hand, within the rotor the structure was uniform. These differences in structure are apparently primarily the result of heat treatment. It was also found that the structure along the periphery of the disk was not uniform, and in both rotors transition phases were found in the metallographic structure of the metal. On the other hand, in rotor III the structure was found to be uniform and the differences between the periphery and the central portion extremely small. The lamellar structure of the pearlite clearly shows that the piece was cooled uniformly and slowly. The part dealing with the X-ray examination of the material cannot be abstracted here because of lack of space. It may be pointed out, however, that it has given interesting and informative results.

Among other things it has been found that carbon was nonuniformly distributed along the length of the axis and the heat treatment was apparently non-uniform, both in the longitudinal direction and in the direction of depth. In the piece regions of different structure and different hardness were found irregularly distributed through the entire mass. Such regions produce differences in the coefficient of expansion which can be clearly proved.

Considering the high operating temperatures as well as the fact that structural irregularities of the metal may be located on one side of the forging only, it would appear that the cooperation of these factors may lead to a bending of the rotor. To prevent this it is not only necessary to select the proper materials but also to take proper precautions in the manufacture and operation of the turbine. In view of this the insurance company jointly with the manufacturer of the forgings, the manufacturer of the turbine, and the user of the latter have worked out certain recommendations, as follows:

(1) The steel mills are fully capable today of furnishing turbine rotors with an elastic limit up to 36 kg per sq mm (51,192 lb per sq in.) without employing oil quench.

(2) If higher elastic limits than 36 kg per sq mm are required, a material should be used which is free from temper-porosity, such as tungsten or molybdenum steel.

(3) In order to obtain the highest freedom from stresses it is recommended that the elastic limit be selected as low as possible.

(4) Bearing in mind the unavoidable differences in temperature and the bending of the shaft produced thereby, it is recommended that particular attention be paid to clearances in the stuffing box.

(5) Because of this also, heating of the turbine while it is standing still is to be avoided.

(6) In order to avoid rusting of the turbine while it is standing still, it is necessary to keep it dry, that is, to protect it from leakage steam.

(7) If a rotor has been straightened it should be maintained after the straightening operation for at least 24 hr (12 hr for rotors of medium and small size) at a temperature which is, if possible, 50 C (90 F) above the live-steam temperature, providing the material of the rotors, blading, etc., permit it. Temperatures below 400 C (752 deg fahr) require a longer preheating time.

(8) It is recommended that new rotors be subjected to the same treatment as stated in the preceding paragraph. (Dr. of Engrg. Deibholz in cooperation with the Engineering Office of the Allianz und Stuttgarter Verein, Versucherungs-A.G., issue 1, January, 1933, pp. 1–14, and 2 sheets of photomicrographs, *epA*)

High Steam Economy Under Adverse Conditions

THE Lincoln Wharf Station of the Boston Elevated Railway produces a kilowatt on less than a pound of coal with a moderate temperature and pressure and a fluctuating load. Moreover, it is necessary to shut down the unit every night, except during the peak-load period in the winter.

The turbine operates with a pressure of 425 lb gage and a temperature of 725 F at the throttle, and exhausts at an absolute pressure of one inch of mercury. Steam is extracted at four points for regenerative feedwater heating. The generator is rated at 35,000 kw, 90 per cent power factor, 13,200 volts, 25 cycles, 1500 rpm. An older 25,000-kw unit in the same station is used mainly as a stand-by.

The motor-operated turning gear is used to roll the spindle at about 34 rpm when the turbine is not in operation. Oil to the turbine-journal bearings is supplied by a motor-driven auxiliary pump, so that the steam-driven auxiliary oil pump and other auxiliaries are kept idle during shutdowns.

To reduce cylinder distortion caused by non-uniform heating during the starting period, steam from the primary valve (governor controlled) enters the steam belt extending completely around the circumference and heats the cylinder uniformly.

In view of the fluctuating load on the unit the turbine elements, such as nozzles, impulse blading, and reaction blading, have been so proportioned as to produce a flat efficiency curve. This obviates the necessity of operating on base load for good efficiency. The spindle is a single-piece carbon-steel forging. The water catcher is installed at the inlet to the last row of stationary blades. It entrains some of the moisture content of the steam and drains it to the condenser. The effectiveness of this method of inter-stage drainage is shown by the results of tests conducted on this moisture catcher:

Load, kw	17,800	25,500	33,000
Moisture extracted, lb per hr	2,635	3,210	3,500
Moisture extracted, per cent of moisture	20.2	17.0	14.2

The extraction of this moisture improves the efficiency of the last stage of blading and also reduces the erosion on the last row of rotating blading. (R. L. Reynolds, Turbine Engr., Westinghouse Elec. & Mfg. Co., in *The Electric Journal*, vol. 31, no. 7, July, 1934, pp. 269–271, 3 figs., *d*)

The La Mont Steam Boiler in Germany

DURING the last year the technical development of this boiler made considerable progress in Germany. In practical service the typical features of this boiler, that is, forced circulation and positive distribution of the water, have proved to be of advantage. Certain new constructional elements have been introduced in Germany. The object of the positive control of the water distribution in the tube system by means of special nozzles is to prevent water flowing only through the less intensely heated tubes which offer less resistance than the tubes which are exposed to the hottest gases. Fig. 4 of the original paper shows a throttling nozzle of a type which has been quite extensively used. Ahead of the throttle proper there is a strainer which keeps away particles likely to clog the nozzle.

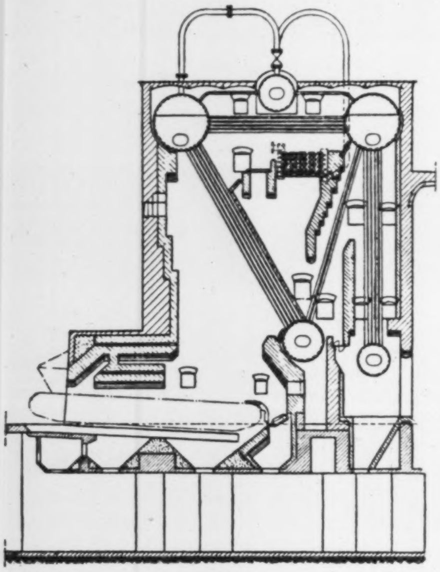


FIG. 8 VERTICAL-TUBE BOILER, 300 SQ M (3200 SQ FT) HEATING SURFACE, OUTPUT 10 TONS OF STEAM PER HOUR

The important element of the La Mont boiler is the circulation pump. As it works against the head of about 35 lb per sq in., a simple single-stage centrifugal pump with overhung impeller may be employed. The power required for driving the pump is from 0.5 to 0.8 per cent of the energy of the steam produced and this amount is made up for by the low radiation losses that are the result of the relatively smaller size of the boiler. In many plants a reserve pump has been dispensed with, as the circulation pump proved to be entirely dependable. For larger plants it has been found useful to provide two pumps which work in parallel but have different drives. In case one pump fails the boiler plant is not in danger.

The first La Mont boilers were waste-heat boilers. The small diameter of the tubes offered a means of efficient utilization of the heat. The tubes in such a boiler may be arranged vertically, horizontally, or at an angle. The possibility of accommodating the heating elements and the steam drum in separate rooms makes the La Mont boiler particularly suitable for the utilization of the heat of the exhaust gases from Diesel engines aboard ship. Several large installations of that kind are cited.

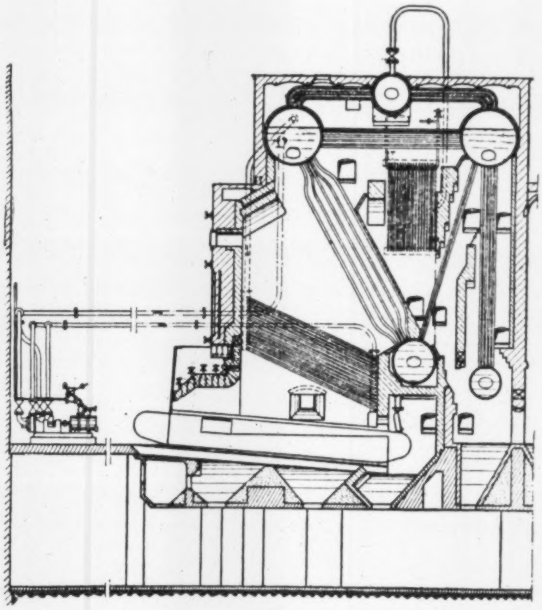


FIG. 9 BOILER OF FIG. 8 RECONSTRUCTED AND EQUIPPED WITH LA MONT COOLING TUBES—OUTPUT INCREASED TO 18 OR 20 TONS PER HOUR

By lining the walls of existing combustion chambers with cooling tubes, the output of the boiler is raised without increasing the load on the boiler proper, while, at the same time, the life of the setting is considerably prolonged. Thanks to the positive circulation, the La Mont system offers the possibility of adding a lining of radiation heating surfaces to the walls of the combustion chamber of existing boilers without any changes in the boiler itself or in the setting. The tubes of this lining may be arranged so close together that they touch each other. By installing such cooling tubes, the output of existing boilers has been raised by from 100 to 200 per cent, a firing plant of correspondingly larger capacity having been provided.

In Germany, the first combustion-chamber lining of this type was installed in May, 1931, by the F. L. Oschatz Co., of Meerane (Saxony), for the Gewerkschaft Deutschland, at Olsnitz, Erzgebirge. At this colliery, there were in operation four boilers with heating surfaces of 300 sq m (3200 sq ft) and an average output of 10 tons of steam per hour (Fig. 8). It was desired to raise the output of steam. The grates of two of the boilers were first lengthened and the front walls were relocated. At the same time, a La Mont tube lining with a heating surface of approximately 55 sq m (593 sq ft) was installed, Fig. 9. By these means, the boiler output could be raised to 18 or 20 tons per hour. A third boiler was provided with a larger grate and a La Mont lining with a surface of 100 sq m (1076 sq ft) and the output increased thereby to 30 tons, or treble the original capacity. This plant, as well as a large number of others of a similar kind, have been in continuous service for years and have given entire satisfaction and proved their reliability.

Instances are cited of the installation of La Mont boilers in connection with other boilers and boilers of higher pressure in existing plants. A number of marine boilers of that type have been introduced. Thus, all the flues of two boilers on the S.S. *Stassfurt* were equipped with La Mont tubes, the output thereby being raised from 27 to 32 tons per hr and the efficiency from 80 to 82 per cent. Although the welded joints of these coils are exposed to the hottest fire, no trouble of any kind has been experienced. In other places complete La Mont marine boilers for oil and coal firing have been installed, and the original article shows in section such a boiler with a flat-grate furnace. It is said that boilers of this type installed in place of the conventional marine cylindrical or water-tube boilers yield a considerably higher output without requiring more room, and make it possible to raise steam of high pressure with its attendant economies. (Dr. of Engrg. Vorkauf, in *Engineering Progress*, vol. 15, no. 2, February, 1934, pp. 35–39, 11 figs., *d*)

High Burning Rates on Underfeed Stokers

THESE results are based on the performance of an experimental installation made by the Philadelphia Electric Co. on its No. 8 boiler at Chester Street to determine the limitations in stoker performance with the various types of coal available in its territory. It is claimed that the tests have shown that coals having characteristics which prevented successful operation on other stokers could be satisfactorily burned with good efficiencies at higher rates than is the usual practise with better grades of coal. The boiler unit is of the Stirling type with a heating surface of 14,217 sq ft and a 50,276-sq ft tubular preheater. It was originally constructed with an entirely refractory furnace, but later the bridgewall and portions of the side walls were rebuilt using bare-steel water-cooled blocks. The experimental stoker is 11 retorts wide and 33 tuyères long with a projected area of 295 sq ft. Manually operated sectionalized air control is provided in three zones. The damper operation is explained in the original article.

The stoker was originally constructed with standard tuyères and with one ram and two pushers per retort. Efforts to operate during the adjustment period with this construction resulted in very excessive burning of tuyères and side plates, so that before experimental operation was begun in the spring of 1932, the two pushers were replaced by a three-pusher arrangement, where the two upper ones were square nosed, each 6 in. high. The third pusher was of the same height but wedge-shaped, so as to provide more lifting action at the lower end of the fuel bed where clinkers would be more difficult to handle. The standard tuyères were placed with a thinner tuyère of a design which brought the incoming air into better contact with all of the tuyère metal and improved diffusion of air to the fuel bed. Both of these changes contributed to a very substantial reduction in burning of tuyères and side plates.

Originally, large deposits of soot were formed in the rear of the boiler and at the bottom of the air preheater. An arrangement was made permitting the collection of this soot and its return to the stoker hopper where it was fed in with the coal and burned. Following the changes made, a definite series of tests was planned with various ratios of air introduced through the zoned air control. The best results were obtained when 15 per cent of the air was introduced through the first zone, 50 per cent through the second, and 35 per cent through the third or extension-grate zone. Analysis has indicated that about 65 per cent of the total air for combustion entered the furnace through these control zones, 15 per cent through the bridgewall air boxes, and 20 per cent by leakage through the stoker.

The object in admitting less air to the A zone was to produce a low burning rate during the coking period when the fuel would have the greatest tendency to blow. This necessitated a much higher burning rate on the area of the grate below the A zone. By the time the fuel had passed the A zone it was thoroughly coked and, therefore, offered much greater resistance to blowing. That this theory is apparently correct was borne out by the test results in which the lowest fly-refuse loss up the stack was obtained during those tests on which the percentage of air is admitted to the A zone was the least, namely, with the ratios previously outlined. Experimental runs also indicated that the best results were obtained with a comparatively heavy fuel bed, about 18 in. deep, and low excess air with a small CO loss. Lighter fires with greater excess air and no CO soon became unmanageable, resulting in very considerable blowing and increase of fly ash in the stack.

With the old stoker the maximum output was about 150,000,000 Btu per hr, which may be considered as a continuous rating since the tests were 24 hr in duration. On the new stoker the maximum output was 250,000,000 Btu per hr, which may be considered excessive as fuel-bed conditions indicated that these ratings would be maintained satisfactorily only for periods of about 4 or 5 hr. At the end of that time there was an excessive amount of smoke emission from the stack. Because of load conditions at the plant it was not possible to run the tests on the new stoker for greater duration than 10 hr, but from the conditions of the fuel bed at the end of the lower rating tests, it is believed that a continuous output of at least 215,000,000 Btu can be maintained with satisfactory operation.

Fly-refuse loss up the stack was determined from a measured sample of the stack gas taken from a vertical rectangular flue between the preheater outlet and the induced-draft fan. The loading of this sample was determined by separating the solid material and filtering the gas through a cloth bag. The solid material was then weighed and analyzed for combustible. The stoker demonstrated its ability to burn some of the cheaper coals carrying ash with low-temperature softening characteristics, which heretofore could not be satisfactorily burned on stokers. The coal selected, a Central Pennsylvania coal, produced clinkers which fused to the tuyères and the side plates of the stoker. Its price range is 25 to 40 per cent lower than that for good stoker coals and it differs from the latter principally in the initial deformation point of the ash which is about 2100 F. In coals which are burned successfully on standard stokers the ash has an initial deformation temperature usually within 50 to 100 deg of the softening or fusion temperature of the ash.

The improvements in the stoker and related apparatus which gave it the ability to burn coal at higher ratings with reasonable efficiencies are said to be the result of the following:

(1) Use of carborundum plates on the extension-grate section.

(2) Use of larger clinker grinders with deeper ashpits having diverging slopes.

(3) Zoned air admission with control and indicated performance on a panel in front of the boiler.

(4) Deeper retorts and better-fitted interior parts.

(5) Improved pusher design with ability to increase or decrease the stroke of each individual pusher from the front of the stoker.

(6) Reclamation and burning of soot which normally is thrown away with the ash.

(7) The introduction of air at the bridgewall which completes the combustion of particles blown to this section of the fuel bed.

(8) The use of slag lips on the side and rear walls to prevent accumulations along the side and back of the fuel bed.

(9) General improvements in mechanical strength of stoker parts. (E. L. Hopping, Engr., Philadelphia Electric Co., in *Modern Power and Engineering*, vol. 28, no. 5, May, 1934, pp. 33–36 and 38, illustrated, *e*)

RAILROAD ENGINEERING

Retardation Controller

THIS is a device to limit the rate of braking to a predetermined value and to keep it within narrow limits during emergency or service stops. It may be used in connection with any air brake or electropneumatic brake and has been recently demonstrated in tests conducted on the Third Avenue Railway in New York City by representatives of the Westinghouse Traction Brake Co. It keeps the air pressure in the brake cylinder within such narrow limits that the rate of retardation is held constant at a predetermined value regardless of the action of the operator in making the brake application unless he desires a lower rate. Its essential feature is a pendulum mounted longitudinally in the car, so that the swing is a function of the retardation rate. It makes two electric contacts in sequence as it is thrown by inertia from the vertical position during retardation. As it swings forward as soon as the brake is applied the first contact is made at the instant when the desired retardation is reached. This acts automatically to move an electric valve to lap position, preventing further entry of the air into the brake cylinder. The pendulum opens the exhaust port of the valve, releasing air from the brake cylinder if the rate of retardation increases beyond a certain limit, and when the retardation rate has been reduced sufficiently a second contact is broken, the exhaust port is closed, and the remaining air is held at the reduced pressure. (*Transit Journal*, vol. 78, no. 3, March, 1934, pp. 88–89, 4 figs., *d*)

The Russian "Ball" Railway

IN THIS new type of railway huge balls containing driving motors supplant the wheels of a train. The permanent way is a concrete trough reinforced with sheet steel. The balls, placed one at each end of every car, have a diameter equal to the car width but are somewhat flattened at both sides. Each ball contains a driving motor. Power is supplied from a split tube overhead, a contact shuttle being pulled in each of the three aluminum tubes by a cable rope from the car roof, feeding power from the alternating-current circuit to the driving motors. The motors are fixed inside the balls which are driven by means of a friction coupling.

During 1932 a trough line 2.85 miles long was constructed near Moscow and five cars were built for experimental purposes. The car body was 2 ft 5 in. wide and was mounted on two steel balls of the same diameter covered with a rubber composition. Each ball was driven by an inside-mounted $1^1/_2$-hp motor through a friction gear, the ball supporting the car body by means of roller-bearing journals. With this model speeds of about 50 mph were attained. A line is now designed for construction between Moscow and Leningrad. The cars will be 12 ft 2 in. wide and 128 ft long and will be mounted on two balls of the same diameter. They will accommodate 110 seated passengers and have a top speed from 160 to 180 mph. Before this line is built, however, a 30-mile experimental line is to be built by the government between Moscow and Noginsk during 1934. For this line six cars were ordered from the Moscow Dynamo Works, each car 82 ft long to seat 50 to 60 passengers. (*Electric Traction and Bus Journal*, vol. 30, no. 2, February, 1934, pp. 45 and 47, 1 fig., *d*)

High-Speed Freight Locomotives in Germany

WHEN, after the termination of the war, the railways of the several German States were merged into the German Railroad Company, standardized constructions were developed for all classes of locomotives, to be used in case of shortages arising in a natural way in the rolling stock.

For hauling heavy freight trains, a 1-E (2-10-0) locomotive was included in this program of standard types. The first locomotives were equipped with three and alternatively with two cylinders for trial purposes. As, by that time, the air brake had been introduced on the freight trains, this 1-E locomotive was designed for a speed of 44 mph and an adhesive weight of 100 tons. The diameter of 55-in. of the coupled wheels was retained. Later only two cylinders were provided.

With a speed of 44 mph the present limit of economic traveling speed for freight trains which have to convey all classes of goods seems to have been reached for countries of the size of Germany. Since there is no telling, however, what train speeds the railways may be compelled to go to by the competition of the motor traffic, measures have been taken, on the one hand, for making the modern freight-train locomotives capable of hauling the trains at 50 mph, for which purpose a slight reduction in the size of the counterweights will be necessary, while, on the other hand, special engine types for this speed were designed.

Since in 1931-1932, exhaustive trials were to be made with a working pressure of 355 lb per sq in. in locomotive boilers of the conventional type, the trial program was made to include also the design and construction of two 1-E four-cylinder compound locomotives for a speed of 50 mph, which are at present undergoing trial. The high pressure made it necessary to let the steam expand in two stages, so that the compound type was the natural result, while, on the other hand, the four-cylinder construction had to be chosen on account of the cylinder dimensions. The excellent balancing properties inherent in the four-cylinder system could, therefore, be used, and increase in the diameter of the driving and coupled wheels avoided. The trials have shown that these considerations were correct.

In order to avoid excessive boiler weights due to the increased steam pressure, a complete boiler shell was made of a heat-resisting carbon steel. The rivets were made of an alloy steel with no less than 0.25 per cent of copper to forestall corrosion and a tensile strength of 48,000 lb per sq in. Preliminary tests have shown that rivets of such relatively soft material when driven in were comparable to a material of 71,000 lb per sq in. tensile strength as regards hardness and strength. The boiler plates showed a tensile strength of 71,000 to 78,000 lb per sq in. and a yield point at 440 F of 40,000 lb per sq in. In building the boiler it was found that boiler plates of the kind selected represent practically the upper limit of cold-workability by bending, fitting, and calking.

In view of the high temperature of the boiler, copper was not considered a suitable material for the firebox, and from the beginning steel was chosen for this part. Extensive trials with low-pressure locomotives equipped with thin-walled fireboxes of steel had, moreover, justified the hope that the problem of the steel firebox could be solved by an appropriate design and careful selection of the materials for the plates and staybolts. Accordingly, both trial locomotives were equipped with fireboxes of steel 0.4 in. thick and with steel staybolts. Since the results of the trials so far had shown that nickel steel was the most suitable material, one of the fireboxes was made of 3-per cent nickel steel, while the other, for the sake of special trial, was made of a carbon steel of 64,000 to 73,000 lb per sq in. tensile strength and a yield point of 45,500 lb per sq in. at 68 F.

The staybolts and crown staybolts are made of the standard "St. C 10.61" steel, a soft material capable of being toughened by heat treatment, but not heat-treated in this instance. The fundamental idea in choosing this material was to avoid the leakages caused by contraction, such as are occasionally observed on steel fireboxes, where hard plates are used with soft staybolts. Since then other trial engines of the 355-lb group had been equipped with staybolts of harder material and fireboxes of relatively soft plates, while in other engines materials of equal hardness were used, and it was hoped that the trials would show what the best combination of materials might be. During those most searching trials, however, leakages developed in the fireboxes of all six trial locomotives, and it will be necessary to make further investigations to enable a safe judgment to be passed on the question whether the Stephenson boiler is suitable for working at a pressure of 355 lb per sq in. The original article gives a table containing the principal data on these high-speed locomotives. (Dr. of Engrg. R. P. Wagner, in *Engineering Progress*, vol. 15, no. 2, February, 1934, pp. 21–23, 3 figs., *d*)

SPECIAL MACHINERY

Remote Control of Valves at Compressor Plants

THIS article deals particularly with the control of valves in gas plants and describes the installation made at two plants in California.

In the larger of the two plants, which is located near Ventura, Calif., there are four separate intake lines, three low-pressure and one high-pressure, which operate under a pressure of from 250 to 300 lb per sq in. In each low-pressure line is installed a butterfly valve operated by an air-actuated piston, and in the high-pressure line there is an emergency stop valve of the swing-check type which is closed by a falling weight released by an air-actuated diaphragm. In two fuel-gas lines balanced-valve-type stop valves are installed operated by falling weights that are released by air-actuated diaphragms. Air supply lines to all valves lead from a central control point located away from the danger zone. Compressed air for operating all valves is supplied through one common valve so that all will operate simultaneously. At this control point is located the plant blow-down valve which is installed in a large line leading to this point and which connects to the plant discharge line just ahead of a non-return valve which prevents the gas from blowing back into the plant in case of a ruptured line. The blow-down valve is of the lubricated-plug type and is hand operated.

In case an emergency arises, an operator opens the stop valves' air supply line, which closes off all intake lines and stops the compressors, and also opens the blow-down valve, which releases all gas in the plant, the non-return valve in the discharge line having closed upon cessation of flow. Tests have shown the plant to be free of gas in $1^1/_2$ min.

A much simpler apparatus is installed in the smaller of the two plants located near Somis, Calif., simpler because the plant is a booster.

In conclusion, the author remarks that although in recent years A.S.M.E. Safety Codes have become more and more conservative, there still remains danger of equipment failure, and in the design of compressor plants equipment similar to that described should be provided. (C. S. Pool, Gen. Supt., Industrial Fuel Supply Co., Los Angeles, Calif., in *Western Gas*, vol. 10, no. 4, April, 1934, pp. 30–32, *d*)

SPECIAL PROCESSES

Manufacture of Nickel Parts by Sintering

THIS process has been developed in the course of the last year. It starts with powder made from nickel-carbonyl by a process patented by the I. G. Dye Industry in Germany. This powder is heated to the sintering point and subsequently forced into the proper shape by the application of pressure.

The nickel powder used for the sintering process contains 99.9 per cent nickel. Slight traces of oxygen and carbon volatilize later as carbon monoxide and carbon dioxide, there being none of such admixtures as copper, arsenic, and sulphur. The nickel-carbonyl powder is placed in a suitably shaped mold of a ceramic material or a heat-resisting alloy and shaken for a short time in order to increase the density as far as possible. The mold is then closed with a cover and heated to about 2300 F in a furnace. During this heating period a reduction of volume of some 60 or 70 per cent takes place while the density increases from about 0.07 to 0.11 lb per cu in. The sintering process is generally finished in about 2 hr and the material is then shaped by pressing, forging, or rolling. This process permits of producing from nickel powder pieces of pure nickel of any shape and size which can be subsequently worked by any of the mechanical finishing processes, such as turning, milling, grinding, etc. It is claimed that owing to the absence of pores and inclusions sintered nickel takes a high polish.

On the other hand, the sintering process offers a simple means of making porous metals for such purposes as diaphragms, filters, etc. By filling the sintering mold with layers of different metal carbonyl powders, composite materials, such as bi-metal strips, can be manufactured. These materials are conspicuous for the perfect joint between the layers. (*Engineering Progress*, vol. 15, no. 5, May, 1934, 4 figs., *d*)

STEAM ENGINEERING

Shape and Size of Steam Bubbles

THE shape and size of steam bubbles forming on a heated surface are determined by the surface tension at the boundary between the liquid and the vapor. The differential equation for this boundary surface is

$$S\left(\frac{1}{R_1}+\frac{1}{R_2}\right)+\sigma\varphi = \text{constant}\ldots\ldots\ldots\ldots[1]$$

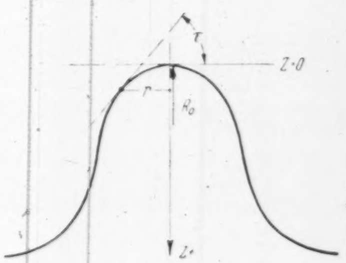


FIG. 10 SHAPE OF THE BOUNDARY SURFACE BETWEEN LIQUID AND STEAM

where R_1 and R_2 are the main radii of curvature of the surfaces at the point under consideration; S is surface tension; σ the density of the liquid; and φ the potential of the external forces expressed in terms of units of mass.

For a body of rotation where gravity is the only external force acting thereon, the following equation holds good:

$$\frac{d \sin \tau}{dr} + \frac{\sin \tau}{r} = -\frac{\sigma g z}{S} + \frac{2}{R_0} = -kz + \frac{2}{R_0} \ldots\ldots [2]$$

Here z is the vertical axis and its point of intersection with the free surface is the zero point; R_0 is the radius of curvature at the zero point, Fig. 10.

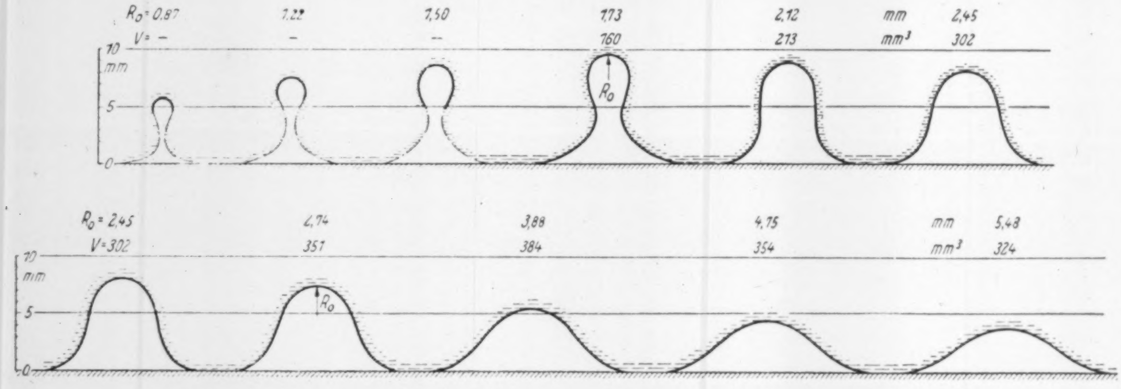


FIG. 11 SHAPE AND CONTENT OF BUBBLES HAVING VARIOUS CURVATURES AT THEIR HIGHEST POINT

Differential Equation [1] has hitherto been solved for particular cases only. It was solved by mechanical quadrature for lying and hanging drops by Bashforth and Adams for a series of magnitudes of the boundary surfaces. The equation for steam bubbles forming on a horizontal heated surface is similar to that for drops hanging from a surface, while, on the other hand, the same relations hold for lying drops and for bubbles located below the horizontal surface. The two groups differ only by the sign of factor k in Equation [2]. For lying drops $+ kz$ should be written on the right-hand side.

Fig. 11 shows a number of bubbles drawn to scale for various radii of curvature at the zero point, these radii determining the shape and size of the bubble. Unfortunately it proved impossible because of lack of fundamental data to determine bubbles smaller than those shown in Fig. 2, and three of those shown could be determined only in so far as the upper part is concerned. The broken-line lower sections are estimates only.

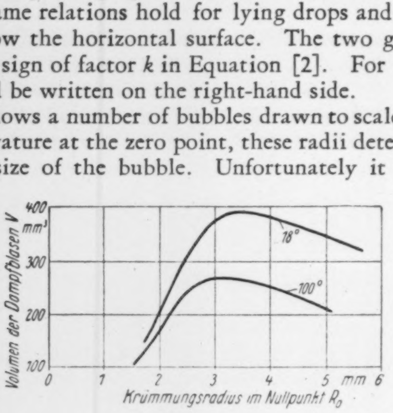


FIG. 12 VOLUMES OF STEAM BUBBLES AT DIFFERENT TEMPERATURES
(Ordinates, volumes of steam bubbles, V; abscissas, radii of curvature at zero point R_0.)

The value of $S = 7.5$ mg per mm which applies to the case of water at 18 C has been selected for the surface tension. This surface tension decreases with increase in temperature and hence the size of the bubbles decreases. Bubbles in liquids other than water may be bigger or smaller, depending on the surface tension.

The bending is greatest at the tip of the bubble where the radius of curvature is least. Going down below it becomes smaller, passing through a zero value into a negative value. In the plane in which the curvature is zero the pressures in the water and in the steam are the same. If it be assumed that the steam pressure remains constant in the entire bubble, the greater water pressure below that plane would flatten the bubble, unless opposed by the counteracting resultant of the surface tension. Contrariwise, the pressure in the water above the plane is smaller. Because of this, equilibrium is achieved by the resultant of surface tension acting inwardly on the steam bubble.

The volume of the bubbles increases with the radius of curvature R_0 and reaches a maximum for $R_0 = 3.5$ mm, becoming smaller again thereafter. To this maximum corresponds the largest possible size of the steam bubble. It is easy to see that this maximum size cannot be exceeded. Thus, for example, for a bubble geometrically similar but twice the size, the static pressures or pressure differences also become twice as great, while the curvature and the forces determined thereby are only half as great. The result is that the bubble is crushed.

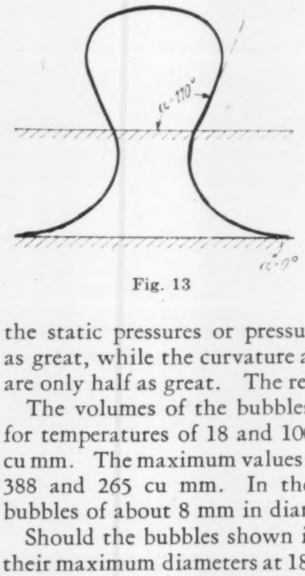


Fig. 13

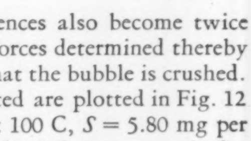


Fig. 14

FIG. 13 INFLUENCE OF BOUNDARY ANGLE ON BUBBLE SIZE

FIG. 14 FORMATION OF BUBBLES ON ROUGH HEATED SURFACES

The volumes of the bubbles generated are plotted in Fig. 12 for temperatures of 18 and 100 C. At 100 C, $S = 5.80$ mg per cu mm. The maximum values for 18 and 100 C, are, respectively, 388 and 265 cu mm. In their tests Jakob and Fritz found bubbles of about 8 mm in diameter on non-wetted surfaces.

Should the bubbles shown in Fig. 11 assume spherical shapes their maximum diameters at 18 and 100 C would be, respectively, 9.1 and 8.0 mm. Here the agreement between tests and calculations is excellent.

It has been assumed hitherto that the boundary angle, which the enveloping surface of the bubble forms at the contact with the heated surface, is zero. Actually, this is usually not so. In the tests made by Jakob and Fritz the value of the angle which was zero was attained by spraying the heated surface with oil. Thus, for $\alpha \approx 110$ deg the bubble (Fig. 13) is shown to consist of only the upper part of the surface plotted.

In conformity with the values found by Jakob and Fritz for half-wetted heated surfaces the volume of the bubbles is correspondingly smaller. Here the angle α between the water and glass is from about 150 to 170 deg. This angle is found in the case of none of the bubbles in Fig. 11, where the largest angle amounts to about 120 deg. No steam bubbles should have been formed under these conditions and the following tentative explanation is given for the fact that they were.

The glass surface is not perfectly smooth, but consists, as has been shown in exaggerated fashion in Fig. 14, of peaks and hollows. In the hollows the steam bubbles can form the angle required; and it is only at such places, or at places where, because of the peculiar state of the surface, such as the presence of impurities, can there be present angles other than those characteristic of the entire surface. As a matter of fact, it can be established that bubbles are always originated at the same places of the heated surfaces. A test has shown that the number of the bubbles formed on a shaving blade dropped into a glass vessel was greater than the number of bubbles on the bottom of the glass vessel. Because of the materially smaller boundary angle between steel and water (in the present case about 110 deg) the bubbles had much more frequent opportunities to form there, with the results that a thick jacket of bubbles formed on the blade.

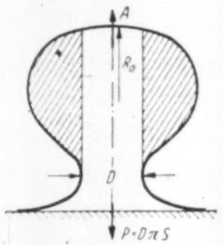


FIG. 15

In accordance with this Jakob and Fritz also found that heated surfaces which are not wetted are covered with a thick layer of closely aggregated steam bubbles.

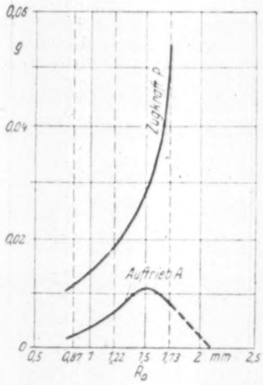


FIG. 16 UPWARD DRIVE AND TENSILE STRAIN AT VARIOUS PITCHES OF BUBBLES AT 18 C (*Auftrieb* = upward drive; *zugkraft* = tensile strain.)

The question as to whether the upward tendency leads to the separation of the bubbles from the bottom of the vessel may be answered by a statement that as a rule the upward force has no action here as shown by the shape of the bubble. It acts only where a contraction of the bubble is produced, that is, when the angle τ at some places becomes greater than 90 deg. Even then the upward force is determined not by the volume of the entire bubble but the volume of the body of rotation shown in the shaded cross-section in Fig. 15.

To determine whether or not the surface tension is sufficient to hold the upward force in equilibrium, it will be assumed for the sake of simplification that the upper part of the bubble is spherical. The upward driving force is then

$$A \sim \left(\frac{4}{3}\pi R_0^3 - \frac{\pi D^2}{4} 2R_0\right)\gamma \ldots\ldots\ldots\ldots [3]$$

and the tensile stress in the narrowest cross-section is

$$P \sim D\pi S \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [4]$$

if one imagines that the action of the surface tension is replaced by an elastic skin surrounding the bubble. As shown for a temperature of 18 C in Fig. 16, the upward forces are always materially smaller than the tension forces so that their effect under the conditions described is not sufficient to tear the bubble away from the surface. (Otto Klüsener, in *Forschung auf dem Gebiete des Ingenieurwesens*, vol. 5, no. 3, issue A, May-June, 1934, pp. 118–120, 7 figs., *te*)

THERMODYNAMICS

Heat Cycles and Their Relative Efficiencies

IN THE April, 1934, issue of *Combustion*, F. H. Rosencrants published an article entitled "Heat Cycles and Their Relative Efficiencies." Some of the discussions which this article provoked are here abstracted.

A. G. Christie, of The Johns Hopkins University, says that based on the experience of the Detroit Edison Co., with a steam temperature of 1000 F, the inlet pressure may be raised to 1200 lb per sq in. abs, which, with five stages of bleeding and no interstage reheating, indicated a possible performance of about 10,900 Btu per kwhr under certain efficiency assumptions. There are boiler plants in America which are now operating at higher annual efficiencies than 87 per cent, while abroad efficiencies as high as 91 per cent have been attained. If one can assume a boiler efficiency of 90 per cent, the cycle performance of the 1200-lb, 1000-F plant without reheating, becomes 10,550 Btu per kwhr, or about 0.76 lb per kwhr with 13,700-Btu coal. Such a plant possesses the simplicity of the 800-lb plant with 7 per cent better efficiency. The 1200-lb 1000-F plant is also about 7 per cent more efficient than the 1200-lb 800-F plant with reheat at 800 F.

G. Gleichman, of Siemens-Schuckert Co., of Berlin, Germany, reports the successful operation of a Benson boiler and its turbine, where, instead of throttling the steam to the turbine at low loads, the boiler pressure itself was lowered. In other words, the set operated with constant throttle temperature and variable boiler pressure. The condenser would be designed for a comparatively low boiler pressure and would be overloaded at the higher pressures. This leads to nearly constant volume at the turbine throttle and the weight of steam admitted is nearly proportional to the boiler pressure. There appears to be about the same Rankine heat per pound available at all pressures, assuming a loss of vacuum at the higher loads. Hence, power output is proportional to steam flow and, in turn, to rate of fuel firing. The final governing will be by the control of fuel feed. The turbine efficiency is maintained over a considerable range of load under these conditions.

G. A. Hendrickson, of the Detroit Edison Co., believes that, with average use factor, the economic balance is in favor of higher temperature without reheat, and that this cycle offers the greatest opportunity for further development. Some time may elapse, however, before the full benefits of increased temperature are commercially attainable. In the meantime the great reduction in heat rate immediately available to the reheat cycle at temperatures suitable for present-day commercial materials makes reheating attractive for base-load plants, where a high use factor affords good opportunity to make profits on the greater investment.

Alfred Iddles, of United Engineers and Constructors, Philadelphia, points out the relatively poor efficiencies of the majority of the power stations operating today, because of the use of what might be called antiquated equipment. Practically every operating company is giving consideration to means for improving the efficiency of existing plants in connection with their increases in capacity. This type of installation has been made in a number of cases, and in all probability will be carried further, particularly since pressure and temperature limits have materially increased. (*Combustion*, vol. 5, no. 11, May, 1934, pp. 9–12, *g*)

CORRESPONDENCE

READERS are asked to make the fullest use of this department of "Mechanical Engineering." Contributions particularly welcomed at all times are discussions of papers published in this journal, brief articles of current interest to mechanical engineers, or comments from members of The American Society of Mechanical Engineers on its activities or policies.

Burning Pulverized Lignite

To the Editor:

Colorado apparently has a great many kinds of coal varying from true lignite to anthracite. The Colorado lignite coal used at Valmont[1] is generally classed as sub-bituminous and while similar to lignite in moisture content and heat value on an ash- and moisture-free basis, it also resembles bituminous coal in that it requires about the same quantity of air per 10,000 Btu. This coal is quite hard, particularly when freshly mined, as evidenced by the grindability factor. Drying, however, results in slacking and possibly affects grindability.

It is evident that unusual precautions must be taken with the design of equipment, temperature of coal, and the time and method of storage, in order to prevent spontaneous combustion. These difficulties would largely disappear with the use of direct firing and mill drying. With high-speed mills higher primary air temperature could be used without danger of fire in the mill. Such mills with direct firing are being used successfully in smaller installations with this fuel and the problems encountered do not differ from those found in using eastern bituminous coal, except that imposed by the effect of moisture on ignition. It is found that, due to moisture, ignition is slower than with bituminous coal, but proper design can compensate for this characteristic.

Mr. Throne has made an interesting comment in comparing the heat release per cubic foot at Valmont with the heat release obtainable at sea level. His statement is probably true except for the increased temperature that would be obtained by greater heat release in the same furnace with the same weight of gases and water-tube area for receiving radiant heat. The rate of heat release is a function of the time each particle of coal is in the furnace and, consequently, the carbon loss for a given coal and given percentage of excess air. With less dense air and gas due to the higher elevation, a larger furnace would be required to keep each particle of coal in the furnace the same length of time, assuming the same total heat release and the same weight of air. This reasoning applies equally to the proximity of the flame to the furnace surfaces.

With bituminous coals, the loss due to unburned carbon for a given design of furnace is apparently proportional to heat release regardless of the coal, whether high or low in volatile matter, in ash, or in grindability. This is on the assumption that coal is ground to the proper fineness in any case. Low-volatile bituminous coal needs to be ground more finely than high-volatile coal, but the former in general is easier to grind. It would be interesting if Mr. Throne could give figures on carbon loss as related to heat release to determine if the relation is the same as bituminous coal.

The Weirton Steel Company, of Weirton, W. Va., have six boilers of 70,000-lb steam capacity, each with slag-tap furnace. These furnaces, with their bottoms directly on the ground and at a level with the boiler-room floor, were originally used for blast-furnace-gas firing. Water walls were applied to the side walls of the original furnaces, the front and rear walls remaining all refractory. A powdered-coal burner was placed in the front wall opposite the gas burner which fired under the bottom drum of a Stirling boiler. The molten ash is drawn off through refractory holes at floor level into trenches and propelled by water jets to an outside pit. The first furnace so fired was operated two years before requiring replacement of the original firebrick floor.

Slag-tap furnaces are well adapted to use with low-fusion-ash coal such as Colorado lignite, provided a high average load can be carried on the boiler in order to obtain a proper flow of ash. This high-load requirement, which is mentioned by the author, would, in all probability, prohibit the use of the slag-tap furnace in the majority of boiler plants in the Colorado region.

Mr. Throne's paper definitely indicates that the relatively low-grade Colorado lignite in pulverized form can be utilized with approximately the same efficiency as the highest grade of eastern coal.

Ollison Craig.[2]

Worcester, Mass.

To the Editor:

Experience with Texas lignite at our Comal plant was similar in many respects to that encountered at Valmont and described by Mr. Throne.[1]

It was found that hot air taken from the preheated air ducts could not be used in the drier or mill systems as an aid in drying the coal because of the fires which it started in some parts of the systems. No definite data are available, however, to show the temperature at which ignition took place. The fires occurred only where the lignite dropped out of the circulating-air streams upon horizontal surfaces and where it was exposed to the full effect of the hot air which was at temperatures between 300 and 440 F. The smoldering type of fires mentioned by the author were also found with our Texas lignite.

No fires due to spontaneous combustion came to our attention even though pulverized coal has remained in our bunkers as long as six weeks at a time. It is believed that the higher moisture and the lower oxygen content of our fuel are responsible for this difference between the Texas and Colorado coals.

It was found that the burning equipment supplied with our last boiler installation easily handled pulverized lignite having 33 to 34 per cent moisture, and that drying to a point at which there is danger of fire is unnecessary. In fact, it was found that better station economy could be obtained when moist coal was used.

It is interesting to note that the actual average boiler efficiency obtained with this plant was very close to the potential efficiency calculated by the author and given in his Table 2.

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[1] See "Burning Pulverized Lignite," by R. F. Throne, Mechanical Engineering, June, 1934, pp. 325–329.

[2] Consulting Engineer, Riley Stoker Corp. Mem. A.S.M.E.

The writer would be interested in knowing if any trouble was encountered at the Valmont station due to condensation of moisture in the air preheaters, such as corrosion of the elements or plugging of the passages by ash particles adhering to the moist plate surfaces. The latter trouble was encountered at Comal in the area where the air first touches the plates but was remedied by by-passing enough preheated air back to the suction of the forced-draft fans to keep the temperature on the gas sides of the plates above the dew-point.

R. R. DABNEY.[3]

San Antonio, Texas.

TO THE EDITOR:

The ash of lignite coal such as that used at the Valmont station[1] differs from eastern coals mainly in these respects: (*a*) Low iron, 7 to 10 per cent Fe_2O_3; (*b*) low sulphur; (*c*) high lime, above 25 per cent $CaO + MgO$; (*d*) very little of the sulphur and iron in the pyritic form, usually less than 20 per cent.

The coal ash and slag have a relatively low fusing temperature and the slag is free flowing due mainly to its high lime (CaO), but difficulty is experienced at low ratings.

The grindability of the lignite coals supplied to this plant will be found in Table 1.

TABLE 1

Mine	Town	County	Seam	No. of Samples	Range of grindability	Average grindability
Columbine	Serene	Weld	Laramie	5	38–43	41
Grant	Frederick	Weld	Laramie	14	39–62	50
Morrison	Erie	Weld	Laramie	3	43–45	44
Clayton	Erie	Weld	Laramie	5	43–54	45
Russell	Frederick	Weld	Laramie	1	. . .	50
Industrial	Superior	Boulder	Laramie	1	. . .	47
Monarch	Broomfield	Boulder	Laramie Monarch	2	44–47	46
Puritan	Dacono	Weld	Laramie Puritan	6	44–49	47
State	Erie	Weld	Boulder Valley	1	. . .	58
Imperial	Erie	Weld	Farmer	1	. . .	61

The effect of drying on grindability has been studied in a limited manner and for a sample containing 8.6 per cent surface moisture, 12.7 per cent inherent, and 21.3 per cent total moisture the test results were as follows: Grindability as received, 45; with 2 per cent surface moisture, 49; air dried with zero surface moisture, 51.

Tests were then made by passing heated air at 275, 310, and 400 F through the grindability machine which resulted in coal temperatures of 110, 115, and 140 F, and grindabilities of 43, 43, and 44, respectively. Another sample showed grindability of 42 as received with 22 per cent moisture and 55 dried to 8 per cent moisture.

TABLE 2

Rating, lb steam per hr	Excess air, per cent	Heat liberation, Btu per cu ft per hr	Combustible in flue dust, per cent	Heat loss, per cent
78,000	36.6	8,990	1.35	0.022
138,100	33.3	16,700	0.71	0.016
199,551	32.0	24,900	1.43	0.028
271,461	32.0	34,200	4.03	0.072

These tests seem to show that predrying increases the grindability while grinding in an atmosphere of hot air reduces the grindability. A longer period between the drying and pulverizing might result in larger increases in grindability than shown here. This explains in part why the pulverizer capacity with predried lignite is not greatly below that obtained with eastern coals, whereas the capacity with undried coal does show a reduction.

The author has referred to the free-burning characteristics of lignite coal which is verified by the low combustible content of the flue dust obtained in tests made on the slag-tap furnace with turbulent vertical burners that will be found reported in Table 2.

R. M. HARDGROVE.[4]

New York, N. Y.

TO THE EDITOR:

Mr. Throne[1] mentions that the selection of fuel-burning equipment is governed by the utilization of fuels peculiar to the region in which a station is developed. From this statement we might assume that the one method suitable for lignite firing is the pulverized method. However, in the Dakotas and in Western Canada, lignite-burning installations prior to 1929 were of the chain-grate-stoker type almost exclusively. In 1928, the University of North Dakota was active in experimenting with pulverizing equipment but in that year results did not warrant approval of the pulverized-fuel system for lignite firing. The performance of coal-drier equipment was not satisfactory, and some of the trial pulverizing installations were not encouraging because of clogging of mills. It is interesting, therefore, to learn that, during 1928 and some years previously, the Public Service Company of Colorado was pioneering in the development of lignite-pulverizing installations with satisfactory results. The many problems encountered in this development and their solutions described by the author represent a valuable contribution to the data on combustion systems in the lignite field, particularly as the results obtained were sufficiently satisfactory to warrant adoption of this type of firing in subsequent extensions.

The introduction of mill drying, slag-tap furnaces, and boiler-tube slag screens greatly diminished many of the initial problems, although with the temperature of the fuel in the mill limited to 200 F, and that of the initial drying gas to 375 F, it would seem that the pulverized product from the mill must contain a high percentage of the initial moisture. The author points out in this connection that it is necessary to reduce the moisture only 2 to 5 points from the initial condition of 20 to 28 per cent to prevent clogging and packing in conveyers and feeders. It is apparent, therefore, that the conveyer and feeder equipment must handle pulverized fuel with a moisture content of 15 to 20 per cent. In the writer's experience with bituminous coal, bridging in the bunkers and clogging of feeders are encountered with moisture content as low as 3 per cent. With the high moisture content in the dust carried through the vents, it would seem that the bag filters would clog incessantly.

The heating coils which were provided on the outside of the screw-conveyer equipment, if effective, must increase considerably the temperature of the fuel being transported with attendant increase in the fire hazard in the storage bunkers. Despite these possible dangers the fact that the boilers have operated continuously for 3700 hr and a subsequent extension of the operating period to 4900 hr would indicate that the preparation and the handling of the fuel were thoroughly under control.

The adoption of mill drying and the unit firing system should eliminate many of the problems of preparation cited by

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[3] Assistant Chief Operating Engineer, Comal Power Plant, San Antonio Public Service Co.

[4] Research Engineer, Fuller-Lehigh Co. Mem. A.S.M.E.

Mr. Throne and it would be interesting to learn if the unit system was considered when the last installation was made in 1929.

Regardless of the pulverizing system used, results of the research and experience at the Valmont station during the last nine years indicate that pulverized-lignite installations can be made with the anticipation that performance results will be satisfactory.

JAMES F. MUIR.[5]

Pittsburgh, Pa.

TO THE EDITOR:

The so-called Colorado lignites, with which Mr. Throne's paper[1] deals, have come into distinction because of their proximity to a substantial market centering around the City of Denver. The mining operation is not expensive and there is a short haul to a concentrated market. In comparison with eastern fuels, these coals are not to be regarded as first class, though under the local marketing conditions they do compete successfully on an economic basis and their cost per Btu is pronouncedly less than that of other Colorado coals more remote from the market.

These Northern Colorado lignites have two predominant characteristics: First, their inherent moisture content and second, their volatile content, which merits further discussion. Analyses do not disclose new chemical combinations but there is something in the percentage ratio which renders them unstable.

The volatile content for the lignite field ranges from about thirty to forty-five per cent, with an average of approximately forty. The peculiarity of this volatile content is its varying temperatures of distillation. Practical experiments indicate that the average temperature is low; often comparable with the boiling temperature of water. The distillation process is unstable throughout the entire range, and is occasionally even confused in the laboratory determination for fixed carbon. The average hydrogen content is about six per cent, and this high percentage may contribute to the instability of the volatiles. In their vapor state, these volatiles ignite readily, but unless subjected to adequate temperature until the process of combustion is completed, they may pass off unburned.

Because of the volatile content and its behavior, it was early discovered that this type of coal must be burned in repose. The best recorded results in small plants were obtained with overfeed chain-grate stokers, with the air supply properly regulated. With this variety of stoker it is necessary to prevent distillation of volatiles at insufficient temperature to promote complete combustion and to this end an extended ignition zone must be provided near the front of the chain. At the end of the ignition zone the furnace should open out to constitute the combustion zone with the bridgewall so arranged that its radiant heat will be reflected back into the ignition zone. With this arrangement, as the volatiles distil off, they rise and strike against the ignition arch to be deflected down again into the face of the fuel bed.

It has been found also that sprinkling the coal as it enters the hopper improves the results. The inherent moisture in the coal is high, somewhat over 20 per cent, but sprinkling apparently holds back the distillation of the volatiles until the fuel temperature is high enough to promote ignition and to maintain combustion.

Pulverized-coal firing facilitates the use of these lignite coals which are well adapted to burning in suspension due to their high volatile content. The theory is that the volatiles respond to localized temperature and ignite, and in burning yield up sufficient heat to support the carbon in combustion until its final reduction into carbon dioxide. But here again the character of the volatiles becomes significant. There are many coals of this particular Colorado lignite region where the volatiles are not able to burn up the carbon. This is not so apt to occur in big installations and hot furnaces as it is in the smaller installations where furnace temperatures may be almost anything. A great deal, too, will depend upon the degree of pulverization, it being apparent that the coarser the coal the more of it will fall to the furnace floor as unburned carbon, and this problem suggests a very careful regulation of combustion air. These lignites require, however, less air for complete combustion than do most bituminous coals. Because of the high volatile content they require considerable primary air, but care must be exercised that the velocity at the burner is not sufficiently great to quench flame propagation. Secondary air must be regulated to complete combustion, but less will be required than for bituminous coals.

The author has indicated that great care is required in the preparation of these coals for the storage system of firing. This suggests the use of unit direct-firing systems thus avoiding storage. A further advantage would seem to accrue to the substitution of the impact type of mill. While ball mills are capable of finer ultimate reduction, considerable crushing power is required, and this coal has a cleavage that does not respond readily to the breaking action of the balls. On the other hand the coal shatters readily to impact, involving a great deal less of pulverization power. It is true that this impact action does not always break down crystalline structure and, therefore, is not entirely adaptable to the extreme fineness of grind required in water-cooled furnaces. For smaller refractory-lined installations, however, this type of mill is adaptable and inexpensive of operation.

The so-called lignite coals constitute a peculiar problem of their own, and standard burners intended for other fuels frequently do not give the desired result. If appropriately handled, however, these coals give good results in pulverized form and constitute the most effective and economical fuel in the particular market region which they serve.

DURKIN VAN LAW.[6]

Denver, Colo.

Number of Active Coils in Helical Springs

TO THE EDITOR:

I regret that the rewriting of my paper "Number of Active Coils in Helical Springs," paper RP-56-4, in the Transactions of The American Society of Mechanical Engineers for June, 1934, resulted in certain errors and misunderstandings.

In Fig. 6, α should be angle *AOB* and ϕ angle *AOC*. In Fig. 7, α should be angle *AOC* and ϕ angle *AOB*. Discussions by J. P. Mahaney and A. M. Wahl were based on early transcripts of the paper in which, due to an error in writing, Poisson's ratio was given incorrectly. The correct value, see page 467, is 0.275. In the ninth from the bottom line of column 1, page 475, the correct value of f is $(e_1 - e_2)/2$.

ROBT. F. VOGT.[7]

Milwaukee, Wis.

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[5] Power Engineer, The West Penn Elect. Co.

[6] Consulting Engineer. Mem. A.S.M.E.

[7] Assistant Chief Consulting Engineer, Allis-Chalmers Manufacturing Co. Mem. A.S.M.E.

A.S.M.E. Boiler Code

Interpretations

THE Boiler Code Committee meets monthly for the purpose of considering communications relative to the Boiler Code. Any one desiring information as to the application of the Code is requested to communicate with the Secretary of the Committee, 29 West 39th St., New York, N. Y.

The procedure of the Committee in handling the cases is as follows: All inquiries must be in written form before they are accepted for consideration. Copies are sent by the Secretary of the Committee to all of the members of the Committee. The interpretation, in the form of a reply, is then prepared by the Committee and passed upon at a regular meeting of the Committee. This interpretation is later submitted to the Council of The American Society of Mechanical Engineers for approval, after which it is issued to the inquirer and published in MECHANICAL ENGINEERING.

Below are given records of the interpretations of this Committee in Cases Nos. 775, and 779 to 783, inclusive, as formulated at the meeting of June 29, 1934. They have been approved by the Council. In accordance with established practise, names of inquirers have been omitted.

CASE No. 775

(Annulled)

CASE No. 779

(In the hands of the Committee)

CASE No. 780

(*Special Rule*)

Inquiry: The use of the high-strength copper-alloy material covered in the inquiry in Case No. 715, which is allowed by the reply to that Case a working stress of 9000 lb per sq in., results in too great a weight for a seamless pressure vessel when computed at that stress for use at high-pressure purposes at very low temperatures. Will it be acceptable, under the Code rules, to compute the working pressure of such seamless vessels fitted with bolted covers to operate in low-temperature service on the basis of a stress allowance of 11,000 lb per sq in., provided that any openings in the cylindrical portion of the shell are adequately reinforced under the rules in Par. U-59? It is pointed out that the ultimate strength and the elastic limit of this material is considerably improved at low temperatures.

Reply: In view of the data submitted, it is the opinion of the Committee that the material referred to, if used in seamless vessels fitted with bolted covers with all openings in the shell adequately reinforced, will give safe results at sub-zero temperatures under a working stress of 11,000 lb per sq in.

CASE No. 781

(*Interpretation of Par. U-36*)

Inquiry: In the application of the fifth section of Par. U-36 of the Code, should the required thickness of the head as determined by the head formula be used in determining the amount of reinforcement required, or should the required thickness of a shell as determined by Par. U-20 be used in fixing the amount of head reinforcement required?

Reply: It is the opinion of the Committee that the thickness to be used in applying the requirements of the fifth section of Par. U-36 shall be that of the head when figured in accordance with the requirements of Par. U-36.

CASE No. 782

(*Special Rule*)

Inquiry: Will it be permissible to use firebox-quality steel plate not over $^5/_8$ in. in thickness conforming to Specifications S-1 for Steel Boiler Plate, except that $1^1/_2$ to $2^1/_2$ per cent nickel is added, resulting in physical qualities as follows:

Tensile strength, lb per sq in.........	70,000 to 80,000
Yield point, per sq in., min..........	0.6 ts
Elongation in 8 in., per cent.........	$\frac{1{,}600{,}000}{\text{ts}}$, but not less than 25 per cent

for the firebox sheets of fire-tube boilers provided that the maximum allowable working stress is not greater than 11,000 lb per sq in.?

Reply: It is the opinion of the Committee that the material specified in the inquiry and used as so outlined will meet Code requirements. If the boiler complies in all other respects with the Code requirements, it may be stamped with the Code symbol.

CASE No. 783

(In the hands of the Committee)

The Strandgren Cyclogiro

(*Continued from 537*)

zontal flight is high, whereas in hovering vertically it is low and may even be zero, depending on the variables just cited.

MAXIMUM THEORETICAL THRUST IN STATIONARY FLIGHT

Let us apply the theorems of momentum and kinetic energy to the case of stationary flight, supposing the velocity of the air in the wake to be constant. Two expressions will be obtained, one for the thrust and the other for the power required, containing, respectively, the second and third power of the velocity in the wake. By eliminating the velocity between these expressions the following equation results:

$$T = 0.63\sqrt[3]{(P^2S)} \qquad \ldots\ldots\ldots\ldots [1]$$

where T denotes the thrust in kilograms, P the power in kilogram-meters per second, and S the surface in square meters obtained by multiplying the diameter of the wheel by the span of the blades. This formula, which is independent of the velocity in the wake, gives the ideal thrust T obtained in an ideal fluid with no losses of energy when the projected surface is S and the power supply P.

During controlled tests, a wheel whose blades had an aspect ratio (ratio of span to depth or to chord) of six gave 80 per cent of the ideal thrust, i.e.,

$$T = 0.5\sqrt[3]{(P^2S)}$$

where P is the power at the motor axle. The five or six per cent of transmission loss is thus included in P. In adopting wings with an aspect ratio of ten or twelve, 90 per cent of the ideal thrust would be available. This is, of course, a very remarkable result.

Equation [1], used by M. Margoulis while director of the Eiffel Laboratory, Paris, in his study on hovering propellers and helicopters, shows the mistake made by many persons who foster the illusion of obtaining enormous lift with small overall dimensions and small power supply. In this respect Equation [1] marks the limit never to be obtained. Taking account of transmission losses, the losses of energy that occur in such a real fluid as air, and the fact that the velocity distribution in the wake is not exactly constant, one may venture to say that the efficiency of this wheel attains, if not the actual practical maximum limit, a value almost equal to it.

EXPERIMENTS ON MODELS

Tests on small models began as early as 1925 at the Aerotechnical Institute of St. Cyr, near Paris, under the supervision of its director, M. Toussaint, and continued for several years, using a wheel 6.6 ft in diameter, the main purpose being that of determining the best law of oscillation.

As the author was at that time his own designer and mechanician, M. Toussaint's permission to use the laboratories and machine shop was of very great value. In the test there was used a special electrically driven vehicle running on a railway track one mile in length with provisions for recording diagrams of the lift and the drag.

By means of these tests the effect on the position of the wings produced by eccentricity of the installation were determined. The aerodynamical tests consisted in measuring lift, drag, and power for different ratios of V to U. The results have been expressed in dimensionless factors.

The experimental verification of the stability was made by a special test, a view of which is shown in Fig. 6. The wheel was driven by a small electric motor located in the fuselage, and the wheel model, in order to imitate the conditions of flight, was suspended with three degrees of freedom about its center of gravity. During the test, deviations in different directions were imparted to the system in order to observe its behavior. It proved to be stable and always came back to the normal position.

The gyroscopic effect of the wheel, which has also been investigated mathematically in detail, may be compared to that of a rolling hoop. If a hoop tends to fall to one side, it turns by itself in that direction and this will check the fall as long as the hoop rotates sufficiently. To guide a hoop, every boy uses this principle, inclining the hoop to the side to which he wishes it to turn. In the cyclogiro the same principle may be used: To turn, say, to the right, the pilot has only to incline the aircraft to the right and it will turn by itself in that direction, in a curve corresponding to the degree of a given inclination.

The author has lately secured the cooperation of the well-known French aeronautical concern, the Liore & Olivier Co.

BOOK REVIEWS AND LIBRARY NOTES

Building an Engineering Career

BUILDING AN ENGINEERING CAREER. By C. C. Williams. McGraw-Hill Book Company, Inc., New York, 1934. Cloth, $5^1/_2 \times 8$ in., 247 pp., 51 figs., $2.

REVIEWED BY DEXTER S. KIMBALL[1]

ONE of the more recent features of engineering education is the necessity of informing the prospective engineer as to what the field of modern engineering really is and also as to the economic and social implications of his chosen profession. In a general way this need exists in all professions, but the rapid spread of modern engineering into every nook and corner of our existence and the social and economic problems that have been raised by modern applied science has, in fifty years, pushed engineering from a somewhat obscure profession to a position of supreme importance. Preparation for this profession can no longer be conducted by any haphazard method. In all good engineering colleges today an effort is being made to help the entering student to understand more fully the problem mentioned in the foregoing, and Dean Williams' book is based upon a very comprehensive effort to "orient" and "motivate" the freshman in the institution he serves. The author in his preface states that in addition to offering the student a bird's-eye view of the field of engineering, the course aims also to help him acquire efficient methods of study and to form good mental habits and at the same time arouse and keep alive his interest in what otherwise may be found to be rather uninteresting topics.

The book is divided into three parts: Part 1, education for engineering, deals with vocations and professions, the profession of engineering, the objectives of engineering education, and how to study engineering.

Part 2 is a historical background of engineering, based on materials and structures, machines and manufactures, heat energy, electrical energy, transportation, and communication.

Part 3 is devoted to the achievements in chemical, civil, electrical, and mechanical engineering, and to the social and economic effects of engineering.

Although the text is strictly non-technical, each chapter has a few mathematical problems appended to it. While these problems, mathematically, should be very easy for any student capable of studying freshman mathematics, they are intended, presumably, to be a drill in method and to illustrate engineering problems.

To the reviewer the best section of the book is Part 1 wherein is described what engineering is and wherein is offered much sound advice and counsel to the young engineer as to methods of study and as to the development of mental habits. The author, also, very sagely remarks that "mathematics is the most reliable index of engineering ability." It is at least the best *negative* index. A student may be an excellent mathematician and yet be unfitted to become a good engineer, but without mathematical ability he has little or no hope of success in the higher realms of the engineering profession.

Part 2 is a highly abridged anticipation of much that the student will receive later on in his course. Usually in orientation courses of this character this portion is given by a number of professors, each an expert in his field. The reviewer's experience with composite lecture courses of this character is that to make such a course a success requires more restraint on the part of the lecturers than most of them possess. Each expert usually feels that he has an opportunity to tell all he knows and the result is often overwhelming and confusing to the student. Possibly an outline like Part 2 which the student can read will help to keep the overenthusiastic teacher down to earth.

Part 3 is, of course, inspirational and designed to arouse the student's curiosity and admiration concerning the work of the engineer. Spectacular achievements are described and interesting episodes concerning men and events are related. This part does much to supplement the descriptive material of Part 1.

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[1] Dean, College of Engineering, Cornell University, Ithaca, N. Y. Past-President, A.S.M.E.

The small size of the book should commend it. Dean Williams has condensed much information into 241 pages. It is suggestive, therefore, rather than exhaustive, which is exactly what an orientation course should be. While written primarily for college students, it should be even more useful for boys in high school who are trying to solve the very difficult problem of what to do with their lives.

After the Shutdown

AFTER THE SHUTDOWN—Part I, The Readjustment of Industrial Workers Displaced by Two Plant Shutdowns. By Ewan Clague and Walter J. Couper. Part II, Former L. Candee Workers in the Depression. By E. Wight Bakke. Institute of Human Relations, Yale University, New Haven, Conn., 1934. Cloth, 6 × 9 in., 153 pp., illus., $2.

REVIEWED BY JAMES W. HOOK[2]

IN "After the Shutdown," Messrs. Clague, Couper, and Bakke offer a case study of what happened to a majority of the 2200 workers who were thrown out of work when the plants of the United States Rubber Company in New Haven and Hartford, Conn., were permanently closed in 1929—the former on April 6 and the latter in late August. The study was made under the auspices of the Institute of Human Relations of Yale University.

The plant in Hartford employed male workers almost exclusively. In New Haven female workers exceeded the male by almost two to one. This was because of the nature of the work done in the respective plants. The Hartford plant produced automobile tires (heavy work) which required specialized talent and machine operators. The New Haven plant produced rubber footwear (light work) which permitted an entirely different type of labor.

The object of the survey was to follow the workers after the shutdown of the plants to determine the length of time they were unemployed, the success they had in their first new jobs, the earnings on their new jobs in relation to the old, and other related facts. Messrs. Clague and Couper followed the workers for eleven months, or during the period before the economic depression struck its full force. Mr. Bakke took up the pursuit at that point for the New Haven plant alone and carried it on for an additional two years, or until the middle of 1932.

Of the 1400 employees in Hartford, 1105 were listed for study and 534, or 48 per cent, were actually found. In New Haven, of the 729 listed, 672, or 92 per cent, were found. The wide differences in those found and interviewed in the two plants are, for the most part, accounted for by the difference in the percentage of women workers and by the fact that many of the workers in Hartford had been recruited from textile districts outside the city when business in those districts was declining. These workers were largely single men, many of them French Canadians, who left the city promptly after the shutdown. The women workers in New Haven were nearly all local citizens who did not seek elsewhere for work.

Many other differences were found to exist in the working forces of the two companies, all of which had a bearing upon what happened after the respective shutdowns. The New Haven plant was the oldest rubber-footwear plant in the United States. It had clung tenaciously to old methods and as a result had drawn to itself a large group of long-service employees. The average length of service of its employees amounted to almost ten years. The Hartford plant was not so old. The average service of its workers was slightly more than five years. It was these differences and the opportunity they gave for comparison that helped to make the whole study interesting and important.

These comparisons (and the book is full of them) show that the effect of a plant shutdown is as variable as the four winds. Age of the worker has its bearing. Economic circumstances another. Sex and marital state another. Length of time on the payroll still another. And more important, perhaps, than all else is the state of business at the time of the shutdown. Is it on the up or down grade?

Chapter 3 deals with the families of the dismissed workers for the first year after the shutdown. This is one of the most interesting parts of the book. In New Haven 606, and in Hartford 400, families were investigated. These families were classified as to marital state and dependents, number of wage-earners in each, families owning their own homes, families having borne serious illness or accident since the shutdown, rent paid, insurance coverage, debts, and calls upon charitable institutions for help. These data for the New Haven plant showed that 95 per cent of the dismissed workers were married or had dependents to support. The average number of wage-earners was $2^{1}/_{2}$ per family. Approximately one-third of the families owned their own homes. Serious illness and accident had befallen 25 per cent of the families. Rent averaged $22 per month. Almost 80 per cent carried some life insurance and in 60 per cent of the cases every member of the family was covered. Debts were reported by slightly in excess of 11 per cent, and a bare $3^{1}/_{2}$ per cent had applied for charitable assistance.

These figures go to prove how important is the family group in our American life. The same fact was discovered in the extensive survey of the industrial unemployed of Connecticut by the Unemployment Commission of that state in 1932. To think of the unemployed as an individual is a blunder of the first magnitude. To do so is to overlook the earning power of the family as a unit in some cases, and the burden placed on a single earner to support a family in others. In all instances, the incidence of unemployment merits special case studies if its consequences are to be accurately evaluated. The methods used by Messrs. Clague, Couper, and Bakke will impress students of the subject as being simple and effective.

Another valuable section of the book is Chapter 4 which describes the experience of 525 of the former workers in the New Haven plant in obtaining new work after the plant shutdown. These workers reported having located 861 new jobs. Of the latter 391 were obtained by applying at the plants, while 311 came through the help of friends or relatives. Newspaper advertisements supplied 22, employment offices 11, and political influence 6. The remainder were supplied through miscellaneous sources, including 73 from influence exerted by the shutdown company itself.

These figures, of course, do not tell the whole story. Back of them lie endless wandering from plant to plant, heartaches and discouragement, time-wasting waits for interviews, and the final acceptance of the job itself, often at beginners' rates of pay. It does seem that some method could be devised to make it easier for job hunters and the available job to be brought together, probably through a reorganized Federal employment service which would merit the confidence of both the employer and the employee to an extent much greater than now exists.

It would be hard to find any common factor revealed by the study that one could pick out as something that would apply to every shutdown that may come in the future. Several plausible conjectures may be drawn, however. One is that the major burden of a shutdown falls upon the displaced workers.

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[2] President, The Geometric Tool Company, New Haven, Conn. Assoc-Mem. A.S.M.E.

Another is that readjustment is more difficult for the older than the younger workers. And the third is that a shutdown involving the wholesale dismissal of faithful workers should be accompanied by a dismissal wage to help tide them over a reasonable period while searching for new jobs.

Mr. Bakke had by far the hardest end of the task in so far as arriving at a sound conclusion was concerned. He followed the workers in the New Haven plant for two years, ending in April, 1932. These years were deep depression years, making it difficult to segregate effects of the shutdown from those resulting from dull business.

WHAT THE STUDY SHOWED

The study showed that in none of the years after the shutdown did the average employee who lost his job work more than 67 per cent of the time, nor the average woman more than 71 per cent. The older men were notably handicapped in obtaining steady work, while those men having the best employment records were those between 30 and 44 years of age. The skilled men suffered most from loss of income. The average income of this group in the third year was $34 less than that of the unskilled workmen, reflecting the dearth of skilled-worker tasks. Mr. Bakke showed that from one-quarter to two-fifths of the laid-off women workers accepted jobs in the sewing trades at wages ranging from $2 to $3 less per week than they were receiving when the plant was shut down. Many other women employees for one reason or another had withdrawn from the labor market entirely. Finally, and most important, he showed that only 3 of the 51 who were given pensions at the time of the shutdown were recipients of public or private charity.

These conclusions do, indeed, show what unemployment means to the average worker. They do not, however, prove conclusively that what happened to these men and women was the result of the permanent closing down of the plant. Part undoubtedly did. But until we could know what the depression would have done to employment in the plant had the latter continued to operate, or what would have happened permanently to the displaced workers if general business had remained prosperous during the three years following the shutdown, it is impossible to find much of a connection between the latter and the employment record and economic circumstances of the worker for three years after. A study of the employment and wage records of some rubber-footwear plant that did not permanently shut down for the three-year period following April 6, 1929, to obtain data for comparison with the New Haven experience, would be most interesting and informative.

The book is an excellent analysis of all that happened to the displaced workers in the two plants that were permanently closed. It is clear and easy to read and the tables in the main text and copious appendix are captioned so as to be easily understood. The study was made by men who have distinguished themselves in social and economic science. Mr. Clague was formerly with the Bureau of Labor Statistics and is now the Director of Research of the Community Council of Philadelphia; Mr. Couper was formerly Assistant-Professor of Economics at Yale and later Assistant Commissioner of Labor of Connecticut and is now on the staff of the Emergency Relief Commission of Connecticut; Mr. Bakke is Assistant Professor of the Science of Society at Yale and the author of "The Unemployed Man," and shorter articles on subjects related to unemployment. Collaborating with them was the Institute of Human Relations of Yale University, whose equipment and personnel for research in the broad fields of the social sciences are unexcelled.

Instruments and Accurate Mechanism

THE DESIGN AND USE OF INSTRUMENTS AND ACCURATE MECHANISM. By T. N. Whitehead. Macmillan Company, New York, 1934. Cloth, $5^{3}/_{4} \times 8^{1}/_{2}$ in., 283 pp., 85 figs., $3.50.

REVIEWED BY W. A. SHEWHART[3]

AN INSTRUMENT is here defined as any mechanism whose function is directly dependent on the accuracy with which the component parts achieve their required relationships. The book is written for those who use or design instruments as thus defined, although some of the underlying principles are applicable to mechanisms in general and others are germane to a large class of light machinery and apparatus which has many of the characteristics of an instrument. Typical examples cited by the author are clock makers' precision lathes, carbureters and magnetos of internal-combustion engines, many components of textile machinery, governors, automatic and hand controls of all sorts, and subsidiary parts of printing presses. Obviously, however, this list could be extended indefinitely. Thus, many examples could be cited in the production of accurate mechanisms in the field of communication alone, and progress in an ever-increasing number of fields of industry depends more and more upon the development of designs in which sources of error have been controlled within preassigned limits. Hence the subject matter is of broad engineering interest.

The first half of the book treats of the theory of errors. Engineers have long been more or less familiar with at least some of the many possible applications of the classical theory of errors based largely upon the early work of Laplace and Gauss. Only one short chapter, however, is devoted to this phase of the subject. The present treatment is novel in that it attempts to relate the three important kinds of error to three kinds of faults in elements of design, where an element is here defined as any partial assembly consisting of two piece-parts related to each other as, for example, two enmeshing cogwheels. The three types of faulty elements are: One not having the desired relation, one that even though constrained by either maintaining or restoring agencies is disturbed from the desired relation for a short period of time, and one whose relation is not uniquely fixed by either maintaining or restoring agencies. These faults of design are associated, respectively, with systematic, short-period, and erratic errors in the functioning of the instrument.

It is significant from a design viewpoint that observed data may be corrected for systematic errors if such are known to be present. Similarly, the effect of unbiased random errors may be reduced at pleasure by taking the arithmetic mean of a sufficiently large number of observed values, which number may be derived from customary error theory. There is, however, no known way of correcting data for erratic errors. Hence a key problem in securing accurate functioning of any instrument is the removal of incomplete elements in design, since these constitute the source of erratic errors. To this end the author gives the following three conditions for complete elements: (1) The reduction of friction to its lowest possible value. (2) Restoring agencies whose moments never fall below some previously assigned definite value when any disturbance of their elements occurs. (3) Elements which are what the author terms inherently complete.

The second part of the book considers some general principles of design in relation to the three kinds of faulty elements. In particular, the advantages and disadvantages are considered for a kinematic design or one wherein the piece-parts forming each element of an instrument are constrained by the theoretically minimum number of point contacts, given the degrees of restraint required. The significance of stiffness, isolation, and protection as means of controlling errors is discussed and illustrated. Particular emphasis is laid on attaining the minimum conditions for accuracy and this is illustrated by a discussion of centering errors. Some of the principles to be observed in planning an instrument are illustrated by a somewhat detailed study of several possible types of design of measuring microscopes. The last chapter discusses in a suggestive manner the significance of the human factor in the use of non-automatic instruments from the viewpoint of the design of the instrument to minimize the contribution of the human element to the resultant error.

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[3] Chairman of the Joint Committee on the Development of Statistical Applications in Engineering and Manufacturing and a member of the technical staff of the Bell Telephone Laboratories, New York, N. Y.

The engineer familiar with the classic as well as the modern theory of errors and statistics will not agree with some of the statements made about the applicability of probability theory, particularly in Chapter 3. Furthermore, the introduction of several new terms for old concepts is somewhat disturbing at times. One such is "stiction," which is defined (p. 85) as the force required to unstick two surfaces to differentiate it from the force required in true sliding friction.

The discussion throughout the book is amply illustrated by many examples chosen from the author's experience as a designer and user of instruments. On the whole it represents an important contribution in the field of design and use of accurately functioning mechanisms, particularly in so far as it stresses underlying principles for controlling errors through design. The book should, therefore, appeal to a wide audience of engineers and scientists, and especially to those concerned with a wide variety of problems where time is not permitted for an exhaustive study of any one so that a knowledge of underlying principles becomes of great service.

The author is assistant professor of business administration in the Harvard School of Business Administration and was lately scientific officer to the British Admiralty.

Hazards of Common Refrigerants

THE COMPARATIVE LIFE, FIRE, AND EXPLOSION HAZARDS OF COMMON REFRIGERANTS. By A. H. Nuckolls. Underwriters' Laboratories Report, Miscellaneous Hazard No. 2375, November 13, 1933. Cloth, $8\frac{1}{2} \times 11$ in., 118 pp., 20 figs. (charts, diagrams, and photographs), 29 tables.

THIS investigation, to develop practical and reliable data for refrigerants, was arranged for and financed by Kinetic Chemicals Incorporated. The report covers a description of samples, identification tests, toxicity tests before decomposition, toxicity tests of decomposition products, analytical tests of decomposition products, ignition-temperature tests, limits of flammability tests, explosion pressure-time tests, and flame-propagation speed tests.

Of the 21 substances considered, seven are now found in wide practise; ammonia, carbon dioxide, methyl chloride, and sulfur dioxide, and three refrigerants: dichlorodifluormethane (F12 or freon), methyl formate (GE refrigerator), and methylene chloride (carrene).

It is explained in the report that refrigerating systems have been used extensively in homes, apartments, hotels, and trade establishments for a number of years, often under conditions where little, if any, attention has been given to maintenance. The records indicate that the life and fire hazards have been small and compare exceedingly favorably with those of other common products and appliances extensively used in hotels, and particularly in homes, for heating and cooking. It is stated that the investigation was not occasioned by alarm regarding present-day uses of common refrigerants. It was recognized however, that a clear understanding and recognition of the possible dangers or hazards of any product is the basis of intelligently determining the conditions for its safe use.

Many of the findings contained in the report are based on experiments with guinea pigs. While they are reliable indicators as to concentrations of vapors and gases that may endanger life, it must be borne in mind that the exact relation between the physiological response in animals and that in man to toxic vapors or gases has not been fully established and varies within certain limits.

The report should serve a very useful purpose in the field of refrigeration in that it calls vivid attention to possible effects of leaks or breaks in refrigerating systems and emphasizes the fact that predictable results are based on governed conditions. No one can predict results in unknown concentrations such as may be encountered in accidental breaks, and the flooding of inhabited rooms or compartments with a highly toxic refrigerant. It should stimulate a more careful study of advantages to be gained from the refrigerants available and a balancing of those advantages against the cost of isolation or other methods of rendering the more dangerous gases safe when used in systems directly affecting human life.—R. R. L.

Alloys of Iron and Tungsten

THE ALLOYS OF IRON AND TUNGSTEN. By J. L. Gregg. Published for The Engineering Foundation by the McGraw-Hill Book Company, Inc., New York, 1934. Cloth, $5\frac{3}{4} \times 9$ in., 511 pp., 14 figs., $6.

REVIEWED BY C. E. MACQUIGG[4]

THIS book—the third in a series of monographs—is the second by this author. The first of the series, "Alloys of Iron and Molybdenum" also by Gregg, was followed by "Alloys of Iron and Silicon" under the joint authorship of Greiner, Marsh, and Stoughton. Other monographs are in preparation under the same auspices, namely, the Iron-Alloys Committee of The Engineering Foundation.

The chapter arrangement of this third book is logical, passing from the mineral sources of the metal successively through constitution (from the structural and metallographic viewpoint), the preparation of the commercial alloys, the constitution of the iron-tungsten-carbon alloys, the manufacture of tungsten steels, the characteristics of tungsten steels, the mechanical properties of tungsten steels, the tungsten magnet steels, and is followed by chapters on high-speed steel. The last chapter covers certain special applications of tungsten alloys, as in heat-resisting steels and stellite.

Writing for the information of the mechanical engineer, suffice it to say that the more strictly metallurgical part of the book, particularly the chapters on the constitutional diagrams, is well worth reading, but will probably interest only the specialist or research worker. This is not true, however, for the text on the steels and cutting tools. Here the mechanical engineer will find material well worth his study, and these features alone will warrant its ownership To be specific, the more practical phase of the book is opened by Chapter 7 on the "Characteristics of Tungsten Steels," in which the author starts with a historical review of the work that has been done on the specific aspects of tungsten on the equilibrium relationships in tungsten steels; from this follows the detailed explanation of how the various critical ranges require the appropriate heat treatments, producing the appropriate structures and the mechanical properties. As an example of the data presented in tabular form for quick availability, one finds information on thermal conductivity, thermal expansion, density, corrodibility in various media, spark testing, welding behavior, etc. Mechanical properties are discussed in a separate chapter. As might be expected, the chapter on tungsten magnet steel is extensive, while there are four chapters on die, tool, and high-speed steels.

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[4] Manager, Union Carbide and Carbon Research Laboratory, Long Island City, N. Y.

A final chapter of considerable usefulness to the delver into metallic alloys, takes up the systems of tungsten with cobalt, nickel, and chromium.

A bibliography of between 500 and 600 references, some as late as 1933, attests the scope of the author's studies. An especially helpful feature of this book is that each chapter ends with a summary in which the author seeks to interpret or accent the text as he has understood it. It is hoped that others assaying similar works can use the author's method of presentation.

Books Received in the Library

Engineer's Sketch-Book of Mechanical Movements, Devices, Appliances, Contrivances, and Details. By T. W. Barber. Sixth edition. E. & F. N. Spon, Ltd., London; Engineers' Book Shop, New York, 1934. Cloth, 5 × 9 in., 355 pp., diagrams, $4.50. For nearly fifty years, this collection of mechanical movements has been popular with designers and mechanics. The present issue contains almost three thousand sketches of machines and machine elements of all kinds, classified and indexed.

Gas Engineers' Handbook, prepared by Gas Engineers' Handbook Committee of Pacific Coast Gas Association, San Francisco, Cal. McGraw-Hill Book Co., New York and London, 1934. Cloth, 6 × 9 in., 1017 pp., diagrams, charts, tables, $7.50. This work is intended to provide the gas engineer with a general handbook containing the constants, formulas, and physical, chemical and engineering data that he frequently requires. All phases of the industry are covered, manufacture, testing, measurement, transmission, distribution, and industrial and domestic use. The book is endorsed by the executive board of the American Gas Association. Gas engineers will find it a convenient reference work, adequate for all ordinary needs.

International Acetylene Association. Official Proceedings, 1933. International Acetylene Association, New York, 1934. Cloth, 6 × 9 in., 214 pp., illus., not for sale to general public. Contains papers presented at the meeting, describing uses of oxyacetylene and oxyacetylene welding in industry; committee reports, addresses, etc.

Kinematics of Machines. By G. L. Guillet. Third edition. John Wiley & Sons, New York, 1934. Cloth, 6 × 9 in., 276 pp., illus., diagrams, charts, tables, $3. In preparing this college text the author has kept in mind the knowledge of physics and mathematics to be expected of the sophomore student, and the amount of time usually allotted to the study of kinematics. The book does not attempt to cover the field completely but gives a selection of material of decidedly practical interest. Fundamental mathematical analyses of motions in machines and useful graphical constructions pertaining to them have been given preference.

Kunststoffe. (Ergebnisse der angewandte physikalischen Chemie, Vol. 2, Pt. 3.) By J. Scheiber, edited by M. Le Blanc. Akademische Verlagsgesellschaft, Leipzig, 1934. Paper, 6 × 9 in., pp. 141–228, diagrams, charts, tables, 7.70 rm. An admirable brief review of the constitution and physical and chemical properties of plastics. A useful bibliography is included.

Machine Drawing. By E. F. Tozer and H. A. Rising. McGraw-Hill Book Co., New York and London, 1934. Cloth, 6 × 9 in., 317 pp., illus., diagrams, tables, $3. This book aims to provide a text and problems in accordance with good drafting-room practise, which will bridge the gap between the fundamentals of freshman drawing and advanced treatises on machine design. The problems have the production of a machine from the conception of the idea to its completion.

Mitteilungen aus den Forschungsanstalten, Vol. 3, Part 2, June, 1934, pp. 29–50. V.D.I. Verlag, Berlin. Paper, 8 × 12 in., illus., charts, diagrams, 2.45 rm. This pamphlet contains three reports from the research laboratories of industrial concerns: The influence of transverse boreholes upon the fatigue resistance of chrom-molybdenum steel; combustion phenomena in iron heating stoves; the erection of the Save river suspension bridge at Belgrad.

WHAT'S GOING ON

A.S.M.E. Council Proposes Restatement of Objectives

UNDER the leadership of the Committee on Policies and Budget, the Council of The American Society of Mechanical Engineers is engaged in strengthening and simplifying Society organization. The first step is the statement of the purposes to be accomplished and the objectives to be attained. Then the grouping of purposes and objectives in logical form provides the basis upon which the organization scheme can be built.

The first step was taken at the Council meeting at Purdue University on July 2 when the Council approved a statement for inclusion in the By-Laws. Final action will be taken in December. The statement follows:

OBJECT

The object of the Society to advance mechanical engineering shall be furthered by activities organized:

(1) To enhance the status of the mechanical engineer by

(*a*) Maintaining high technical and cultural standards for entrance to the Society

(*b*) Cooperating with educational institutions in the maintenance of high standards of engineering education

(*c*) Advocating a high standard of ethical practise

(*d*) Aiding in the adoption of high standard of attainment for the granting of the license to practise by the state

(*e*) Fostering among engineering students the study of the history, traditions, ethics, and functions of the engineering profession

(*f*) Encouraging the personal and professional development of engineers

(*g*) Encouraging a high standard of citizenship among engineers.

(2) To advance the theory and practise of mechanical engineering by

(*a*) Holding meetings for the presentation of original papers

(*b*) Encouraging original research and the presentation of reports

(*c*) Publishing papers and reports and disseminating knowledge and experience of value to mechanical engineers

(*d*) Furthering the work of the Engineering Societies' Library, of which the Library of this Society forms a part

(*e*) Encouraging intercourse among engineers of all branches, for the exchange of mutual experience

(*f*) Granting honors, prizes, awards, etc., that will encourage individual development of engineers and stimulate contributions to the advancement of the profession

(*g*) Developing and promulgating standards, codes, formulas, and recommended practises

(*h*) Fostering international cooperation and understanding among engineers.

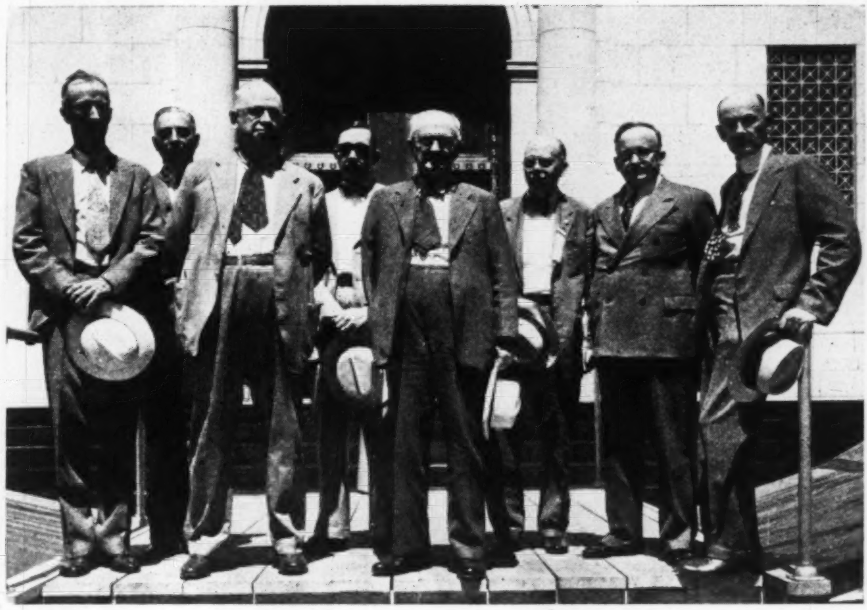
(3) To broaden the usefulness of the mechanical engineering profession in the community by

(*a*) Encouraging the participation of engineers in the solution of public problems, and

(*b*) Cooperating with government in engineering matters.

(4) To support the unity of the engineering profession by

(*a*) Cooperating in the above object with



ENGINEERS RESPONSIBLE FOR THE RECLAMATION BUREAU PAPERS BEING PUBLISHED IN "MECHANICAL ENGINEERING"

(*Left to right:* S. O. Harper, assistant chief engineer; L. N. McClellan, chief electrical engineer; R. F. Walter, chief engineer; I. A. Winter, associate engineer; Dr. Elwood Mead, Commissioner of the U. S. Bureau of Reclamation; H. B. Barnes, chairman of the Denver Committee on Technical Papers; P. A. Kinzie, mechanical engineer; and C. M. Day, chief mechanical engineer.)

other engineering and technical societies, and

(*b*) Seeking in all ways to bring about a well-knit national organization of engineers.

(5) To promote the mutual welfare of the members by

(*a*) Supporting activities looking to the increased employment of engineers and seeking new opportunities for engineering service, and

(*b*) Publicity for engineers and engineering.

This Month's Authors

RALPH E. FLANDERS, who writes on the vexing problem of unemployment, is well known to readers of MECHANICAL ENGINEERING for his writing on economic and engineering subjects. He is rightfully regarded as one of the most lucid and constructive students of industrial problems and has been intimately associated with the NRA as a former member of its Industrial Advisory Board. His leadership in speaking sanely and courageously for industry at the opening of the Code Authority Conference in Washington, March 5, and warning his hearers of the disastrous effects that would follow a universal application of a shorter work week, was a distinct service to his country and to his fellow industrialists. His present paper is another example of his constructive thinking.

James O. McKinsey, analyst of the problems raised by the NRA, is senior partner of James O. McKinsey and Company, accountants and engineers, with offices in Chicago and New York. He is also professor of business policies at the University of Chicago, a director of several companies, member of numerous associations of accountants and management engineers, and author of books on business, management, and accounting.

Phillip A. Kinzie, who writes this month on the butterfly valves for the Boulder power plant, is a native of Denver. He was chief engineer of the Leyner Tractor and Manufacturing Company, of Denver, Colo., when he left, in 1920, to enter the Bureau of Reclamation. At present he is senior engineer in charge of the design of gates and valves in the mechanical-engineering section of the Bureau.

C. B. Strandgren, inventor of the Strandgren cyclogiro, has been developing this unusual type of aircraft for several years.

Herbert S. Ingham, author of the paper on a gear-designing machine, was educated in Los Angeles and was graduated from the California Institute of Technology. Since graduation he has been designing gears for gear-reduction unit assemblies for the Sterling Electric Motors, of Los Angeles, Calif.

W. F. Davidson contributes a brief note on the strength of alloy-steel bolts. He was graduated from the University of Michigan, following which he served as a student and junior engineer with the Westinghouse Electric & Manufacturing Co. After six years of teaching experience at the University of Michigan he became director of research of the Brooklyn Edison Company.

H. B. Holroyd contributes a solution of the problem of calculating the change in the volume of vessels resulting from the deformation of the walls by pressure. He was graduated from Iowa State College in 1924, when he went with the Westinghouse Elec. & Mfg. Co., taking the work in the student course and in the mechanical design school. He was engaged in research work with that company, and in 1926 entered the California Institute of Technology, from which he received the degree of Ph.D. in physics. Since 1929 he has been engaged in research work.

T. G. Estep and D. C. Saylor, who write on the performance of domestic stokers, are members of the faculty of the Carnegie Institute of Technology. Prior to 1913, when he entered the teaching profession, Professor Estep did a considerable amount of testing for the Westinghouse Machine Company, the H. C. Frick Coal Co., and the Babcock and Wilcox Co. Professor Saylor has been associated with the Carnegie Institute of Technology since his graduation in 1920. During this time he has had many contacts with industrial concerns in the making of tests.

"Engineering Index" Changes Hands

IN accordance with a vote of the Council of The American Society of Mechanical Engineers at its meeting in Chicago, Ill., in June, 1933, the Society withdrew its financial support of the Engineering Index on January 1, 1934. A committee, consisting of Collins P. Bliss, Robert M. Gates, and L. C. Morrow, was appointed by President Doty to undertake negotiations with the staff of the former organization which offered to carry on the Index at no financial risk to the Society. The committee found that there was considerable sentiment in favor of the Index, and that the necessary means for saving it could probably be found.

Encouraged by the conclusions of this report, Engineering Index, Inc., was created as a new membership corporation in the State of New York, on June 16, 1934. The incorporators and directors are: Collins P. Bliss, Dean of Engineering, New York University; Ralph E. Flanders, President, Jones and Lamson Machine Company; Robert M. Gates, vice-president, Combustion Engineering Company, Inc.; Clarence F. Hirshfeld, chief, Department of Research, Detroit Edison Company; and L. C. Morrow, editor, *Factory Management and Maintenance*, McGraw-Hill Publishing Company. At the organization meeting of the board of directors, August 6 1934, Collins P. Bliss was elected president of the corporation, L. C. Morrow was elected vice-president, Joshua Eyre Hannum, who has been editor of the Index since 1927, was elected secretary-treasurer, and appointed editor, and Frank Y. Stewart, who has had charge of subscriptions since 1929, was appointed business manager.

At its meeting on July 1, 1934, the Executive Committee of the A.S.M.E. expressed its willingness to lease to this corporation the use of the name *Engineering Index* and the facilities of the former organization.

The newly adopted constitution of the new corporation provides that the corporate membership shall consist of trustees broadly representative of engineering and allied technical societies, educational institutions, and industrial concerns.

It is contemplated that the headquarters of Engineering Index, Inc., will continue to be the Engineering Societies Building, 29 West 39th Street, New York, N. Y., as this location has proved a vital factor in the prompt indexing of the periodicals that come to the Engineering Societies Library.

Fall Meetings of Interest to Engineers

ATTENTION is directed to a number of meetings scheduled for the fall which are likely to be of interest to engineers.

INDUSTRIAL EXTENSION, RUTGERS UNIVERSITY

At Rutgers University, September 6, the Ninth Annual Industrial Extension Conference will be held. Executive training, employee training, and leisure-time programs will be discussed. There will be a morning and an afternoon session, a luncheon, and a dinner.

AMERICAN WELDING SOCIETY

At the Hotel New Yorker, New York, N. Y., October 1 to 5, the American Welding Society holds its fourteenth annual fall meeting. The tentative program lists 30 technical papers and includes a session on fundamental research in welding and a symposium on qualifying operators of welding equipment and expenses incurred therein. The meeting is to open with an address of welcome by the president, Dr. D. S. Jacobus, past-president, A.S.M.E., which will be followed by the presentation of the Miller Memorial Medal Award. The annual banquet will be held Thursday evening, October 4, at the Hotel New Yorker.

NATIONAL METAL CONGRESS

At New York City, The National Metal Congress will be held, October 1 to 6, in the Port Authority Commerce Building, under the auspices of the Society of Metals, with the cooperation of the Metropolitan Section and the Iron and Steel Division of The American Society of Mechanical Engineers. On Tuesday, October 2, at 5:30 p.m., Dr. Zay Jeffries will speak on "Non-Ferrous Metals, Today and Tomorrow," and on Friday, October 5, at 5:30 p.m., R. S. Archer will give a lecture on "Ferrous Metals, Today and Tomorrow." No registration fee will be charged for the lectures, exposition, and sessions. Other societies holding meetings in connection with the congress are the American Institute of Mining and Metallurgical Engineers, the American Welding Society, and the Wire Association. Certificates entitling the holder to a round-trip railway ticket for a fare and a third, from September 27, returning on or before November 1, are available, and may be obtained by addressing the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y.

N.C.S.B.E.E.

At Portland, Ore., October 3 to 6, the National Council of State Boards of Engineering Examiners will hold its fifteenth annual convention with headquarters at the Hotel Multnomah. Among the business to be transacted will be reports of committees on accredited engineering schools, uniform examinations for registration, and the Engineers' Council for Professional Development. All sessions are open to the public.

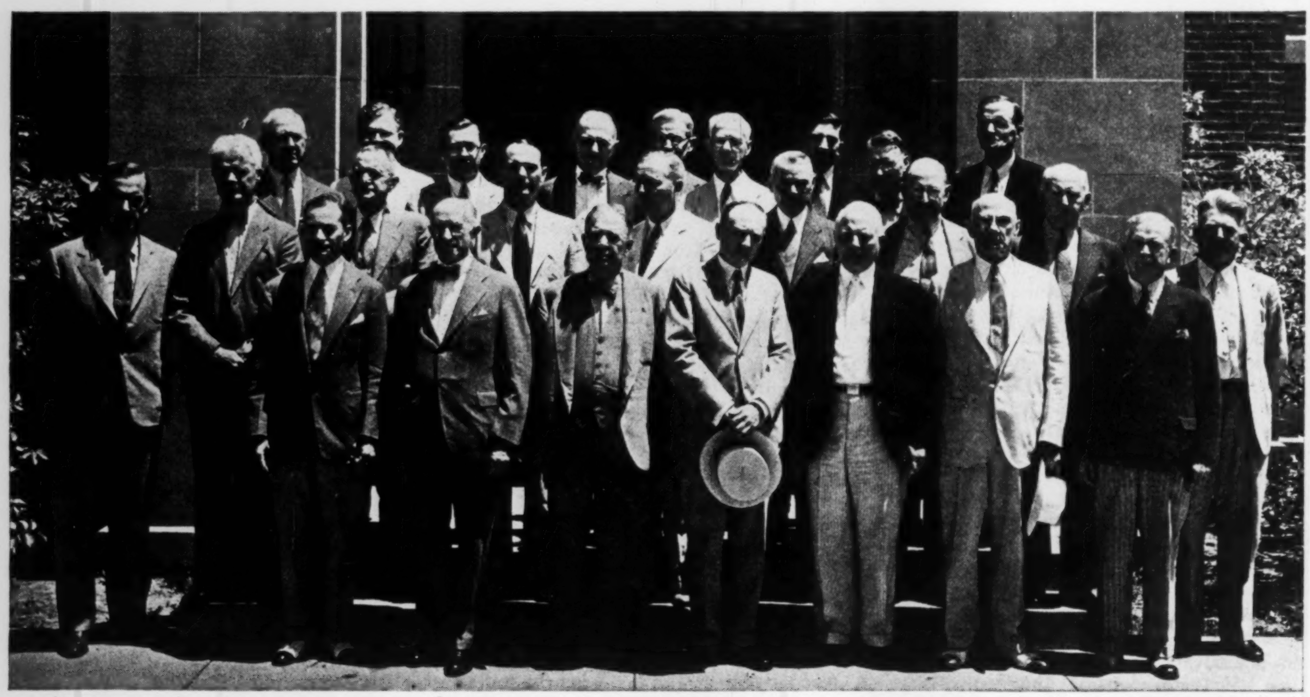
AMERICAN FOUNDRYMEN'S ASSOCIATION

At Philadelphia, Pa., October 22 to 26, the American Foundrymen's Association will hold its 38th annual convention and exposition in connection with the Fifth International Congress. Modern foundry developments in technical knowledge, operating practise, and foundry equipment will be presented. Sessions cover materials handling, refractories, sand, apprentice training, cast iron, steel, malleable cast iron, and non-ferrous castings. The exhibits will open daily 9 a.m. to 5:30 p.m. except Wednesday, when the closing hour will be 10 p.m.

A.S.T.M.-A.S.M.E. High-Temperature Program

THE Joint Research Committee of the A.S.T.M. and A.S.M.E. on Effect of Temperature on the Properties of Metals, at its June meeting, voted to draw up a definite plan, for submission to letter ballot, for experimental study of methods of evaluating the load-carrying ability of metals under prolonged stress at high temperatures. The "creep" of selected materials will be studied by test runs to be carried out, not merely for 1000 or 2000 hours, but for three years, or say 25,000 hours, in order that the validity of various methods of approximation and extrapolation and the relation of various "short-cut" methods to actual long-time performance may be evaluated.

It also voted to study experimentally the progressive changes in toughness, during long exposure to high temperatures, of the 18-8 stainless steel whose creep properties have already been studied in detail by the committee and which were reported in the A.S.M.E.



MEMBERS OF A.S.M.E. COUNCIL AND REPRESENTATIVES OF STANDING COMMITTEES IN ATTENDANCE AT THE COUNCIL MEETING, PURDUE UNIVERSITY, JULY, 1934

Transactions for July, 1934. This continuation of the prior work on this steel is aimed to throw light upon the fact that this lot of material, though free from added "stabilizing" elements, was not embrittled by long sojourn under load at high temperatures in the creep tests. The question whether this austenitic steel passed through a brittle stage, without evidence in the creep curves of any change in load-carrying ability, and then recovered from its brittle state, or whether it never did become embrittled, is of both theoretical and industrial importance, and a material whose other properties have been so thoroughly studied as is the case with this steel, is especially suited for such a study.

Besides these projects the subcommittee on oil-refinery problems is working out an acceptable plan for procurement of experimental and operating information on still tubes for the oil-cracking industry. Another activity is the correlation of data on properties of metals at sub-zero temperatures. A similar correlation of data on seizure will be undertaken, i.e., the effect of temperature on metal-to-metal wear, galling, and sticking. Users and makers of valves that operate at elevated temperatures find this an important problem in which available information is chiefly of an empirical nature. It is hoped to aid in the transition from empiricism toward a sounder basis for engineering design and selection of materials.

H. J. French, chairman of the joint committee, announced the appointment and the acceptance of Major R. A. Bull as chairman of the subcommittee of the main committee on finance. Major Bull's subcommittee is charged with the procurement of funds for the carrying out of the experimental program on fundamentals that are of vital interest to all industries which use metals at high or low temperatures, and to those that produce such metals. More specialized projects with programs of special commercial interest to a particular branch of industry are to be financed apart from the fundamental program. An example of such a specialized interest is that of the oil industry, and its suppliers, in still-tube materials, which is in the hands of a special subcommittee created for that specific purpose and which has its own finance committee headed by H. J. Kerr.

S.A.E.-A.S.M.E. Diesel Fuel Report Ready

A SPECIAL mimeographed report of the S.A.E.-A.S.M.E. Diesel Fuel Research Committee is now ready and may be procured from the secretary of the committee, M. J. Reed, 2 West 45th Street, New York, N. Y., for fifty cents per copy. The report contains the following:

(1) A rough survey of the present fuel-oil market, containing estimates of the amount of fuel oil now being used with a tentative indication of what may be expected several years hence.

(2) History of past efforts toward standardization and the present status of builders' specifications.

(3) A discussion of ignition characteristics with descriptions of the following: (*a*) Crucible and bomb methods for finding self-ignition temperatures; (*b*) engine test methods by critical-compression-ratio method and ignition-lag method; (*c*) description of testing engine; and (*d*) discussion on the standardization of engine-test conditions.

Appendixes provide a detailed history of researches conducted on ignition characteristics. The account covers the researches of Le Mesurier and Stansfield, Garrish and Voss, Boerlage and Broeze, the N.A.C.A., and Pope and Murdock. Under these appendixes the results obtained by these investigators are interpreted for the light thrown upon such factors as injection timing, compression ratio, intake air temperature, jacket temperature, spray characteristics, shape of combustion chamber, engine speed, engine load, and fuel temperature.

Coming A.S.M.E. Meetings

PRINTING INDUSTRIES, PHILADELPHIA, PA., OCT. 8 AND 9

THE Fifth Conference of the Technical Experts in the Printing Industry, under the auspices of the Printing Industries Division of The American Society of Mechanical Engineers will meet at the Benjamin Franklin Hotel, Philadelphia, Pa., October 8 and 9. The program will consist of four sessions, an informal dinner, a luncheon meeting, and visits to two Philadelphia plants. Among the subjects to be discussed are photoengraving, rollers, plates, offset printing, type design, color rotogravure printing, and the aquatone process. A more complete program will be published later.

WOOD INDUSTRIES, CAMDEN, N. J., OCT. 23 AND 24

A meeting under the auspices of the Wood Industries Division of The American Society of Mechanical Engineers will be held at the Hotel Walt Whitman, Camden, N. J., October 23 and 24. The main topic of discussion will be the economics of manufacturing in the wood industries. At one session the economics of replacing manufacturing equipment will be discussed, and another session will be devoted to the reduction of plant overhead. A third session will cover economic lot sizes in the manufacturing of wood products. Inspection trips to local wood-working plants have been planned. A more complete program will be published later.

ANNUAL MEETING, NEW YORK, N. Y., DEC. 3 TO 7

The technical program for the 55th Annual Meeting of The American Society of Mechanical Engineers, to be held in New York City, December 3 to 7, is well under way. Among the sessions being planned are two under the sponsorship of the Fuels Division, one of which will be a symposium on domestic heating. The Power Division is planning a session on industrial power and one on central-station power. The Process Industries Division is holding two sessions on heat transfer, covering liquid-to-liquid heat transfer and evaporation and cooling. The Applied Mechanics Division will sponsor four sessions devoted to vibration, materials, stress analysis, and bearing analysis. It will also cooperate in the session on aeronautics, at which aerodynamics will be the general subject and will re-present two papers presented at the International Congress of Applied Mechanics, held at Cambridge, England, July, 1934.

The Machine Shop Practice Division is planning four sessions, one on machine design, and two research sessions on metal cutting, broaching, and welding. Papers will be presented on education and training in the industries. The Management Division is planning three sessions. A symposium on the engineering aspects of the prevention of occupational disease disability has been planned by the A.S.M.E. Safety Committee. Under the auspices of the Power Test Code Committee, papers will be presented by the subcommittee on fan design, and four other research sessions, covering steam research, mechanical springs, boiler feedwater, and fluid meters have been planned. Additional details will be announced as they become available.

International Steam Table Conference to Be Held Here in September

THE Third International Steam Table Conference will be held in the United States at Washington, D. C., Cambridge, Mass., and New York, N. Y., September 17 to 22, 1934. The first conference was held in London, July, 1929, and the second in Berlin, June, 1930.

This year's conference will be attended by the experts on the thermal properties of steam who are at work in a number of the European countries as well as those in the United States.

The purpose of these conferences is to provide an opportunity for informal comparisons and coordinations of experimental data. At the London conference a skeleton steam table was formulated with a tolerance for each term large enough to include the data of all the laboratories. This table was extended somewhat at the Berlin conference and the amount of the tolerance in many cases was reduced by general agreement. The new steam tables formulated and published by Callendar (England), Hausen (Germany), and Keenan (United States) are based on this skeleton table.

During the four years which have elapsed since the second conference the experimental work has continued in England, Germany, Czechoslovakia, and the United States so it is expected that the third conference will record still further advance toward international agreement in this field.

The coming conference will be sponsored by The American Society of Mechanical Engineers functioning through its Special Research Committee on the Thermal Properties of Steam. It is this committee which has directed and financed the steam research in this country, a study which was begun in June, 1921, and completed in January, 1934, at a cost in excess of $175,000.

The delegates will first inspect the American